MACHINE DESIGN

November 1952

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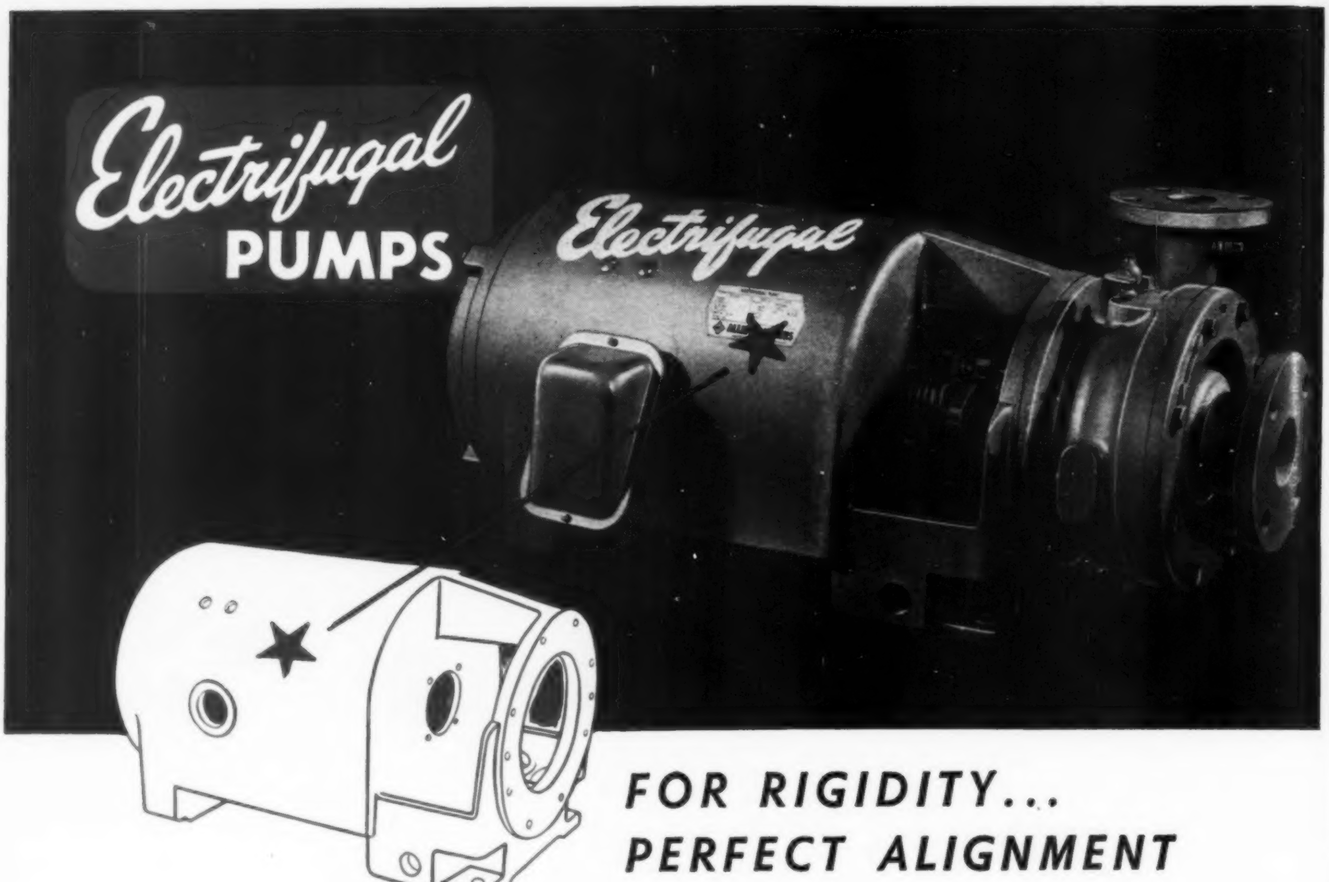

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Buy the Pump Unit with the 1-PIECE FRAME!

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- **No extra parts to buy.**
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FRAME IS OF RIGID cast iron construction with no joints from motor end housing to pump casing.

On the other hand, close-coupled doesn't mean "cramped." There's plenty of space between pump and motor for packing maintenance or replacement.

In addition, wearing rings and shaft sleeves protect casing and shaft from wear and are easily replaced at low cost when worn. As a result you will find maintenance costs very low.

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MACHINE DESIGN

November 1952, Vol. 24—No. 11

THE PROFESSIONAL JOURNAL FOR ENGINEERS AND DESIGNERS

Over the Board

Look Ma, No Jets

In spite of all the talk about jets and the atomic age, the world's work still relies on such old-fashioned principles as piston engines and chemical reactions. A recent swing around the country by the Editor, with stopovers in Denver, Seattle, San Francisco, Los Angeles and back to Cleveland was a marvel of speed and (almost) clock-like timekeeping—all done with piston engines both in the DC-6's and in the airport limousines. The experience prompts the thought that whereas jets could have cut the flight time they couldn't have influenced the ground time—especially waiting for baggage. Seems like the engineers are too many jumps ahead of the rest of the world.

"A New Door on American Thinking"

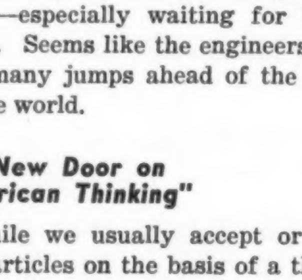
While we usually accept or reject articles on the basis of a thorough review by the editors, occasionally we feel that the opinion of a recognized authority, intimately familiar with a particular field, would be of value. Such was the case with Dr. Eugene Radzimovsky's article, "Bolt Design for Repeated Loading" on Page 135. We thought you might be interested in this comment, from a nationally-known chief engineer of one of the larger fastener manufacturers: "His work is basic and needed by practical men like myself to build on for practical application. . . . I certainly hope you can publish this article since its background of Russian and German engineering opens a new door on American thinking. While some of his thinking contradicts our established line of thought, it is mainly supplementary and, therefore, particularly valuable."

Clutches

Tony Gagne's extensive article on "Clutches" which appeared in our August issue as a special section has attracted many favorable comments and requests for additional copies. Previous experiences in running out of print on popular articles have taught us the wisdom of raising our sights on expected demand. Consequently, for once, we have a good supply on hand. Copies are yours for the asking.

This Month's Cover

Usually our cover pictures illustrate or symbolize an article within the same issue. Occasionally, however, a picture so striking, so appropriate to our field and so well adapted to our cover design comes to our attention that we just can't resist sharing our pleasure in it. This month's cover is such a one. The three jet bombers seen against the deep blue of the substratosphere are Boeing B-47's, symbolizing the complexity and high performance of today's machines.



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Published on the seventh of each month. Subscription in the United States and possessions, Canada, Cuba, Mexico, Central and South America: One year $10. Single copies, $1.00. Other countries one year, $15. Copyright 1952 by The Penton Publishing Company. Acceptance under Act of June 5, 1934. Authorized July 20, 1934.

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ITEMIZED INDEX

Classified for convenience when studying specific design problems

MACHINE DESIGN is indexed in Industrial Arts and Engineering Index Service, both available in libraries generally.

NOVEMBER 1952

It Could Be Sabotage!

THE tale is told of a small job-shop subcontractor for certain aircraft parts who told the inspector not to bother him with tolerances, because every dimension would be "on the nose." The sequel is that the parts he produced were off as much as a tenth of an inch.

If the attitude reflected in this story seems slightly naive, it is not more so than the apparent attitudes of too many design engineers when it comes to specifying tolerances. Of all the phases of a design engineer's work none are as little understood as tolerances.

In spite of all that has been said and is being said on the importance of specifying realistic tolerances, evidence is abundant that the trend is toward closer and closer limits. Some precision parts for functional reasons actually do require such accuracy, and no one will argue the necessity of maintaining them. But when tolerances measured in tenths of a thousandth are being specified for dimensions where one sixty-fourth would be close enough, the situation becomes fraught with serious danger.

Suppose that those wishing to sabotage our military production effort were to plant agents in key positions in engineering departments of the military establishment and elsewhere or were to seduce disgruntled engineers already in such positions. What more effective and insidious method could they use than to see that impossibly close tolerances are specified on certain critical dimensions of certain critical parts? If such specifications resulted in a rejection rate of 50 per cent, or even 25 per cent, the effect on military production could be disastrous.

The idea of such a plot against the United States may seem fantastic, but the present trend if not reversed could have consequences just as fatal. The only solution is for every design engineer to be on the alert and to question critically each tolerance specification, comparing it with the natural, easily maintained tolerance for the process by which the part is to be produced. Our very existence as a free nation may depend upon it.

Colin Carmichael

EDITOR

Developing Engineering Efficiency

Improved effectiveness and productivity in design operations are desirable and well within reach. Orientation of new men and streamlining of staff projects as outlined in this article are real answers to the shortage of engineers

By Arnold R. Smith

Staff Engineer
Office of Graham W. Parker
New York

MUCH has been said and recorded concerning the apparent shortage of engineers in recent years. Few, if any, immediate solutions for the problem have been offered. But out of all that has been written and spoken, one happy note has emerged—recognition has finally been given to the possibilities of increasing the output of the existing force by time-proved methods of management, *Fig.* 1.

Let us start with the problem of the new man. He may either be experienced or a novice, but in either case a period of training is necessary before he becomes familiar with the particular requirements and techniques of the organization. Much of what he has to learn is in the minds of the existing group of engineers or it may be loosely filed somewhere in the office. He may even have to cope with that paradox—the group leader who has been around for twenty years or more and is indispensable because of the extent of his knowledge, but whose concentration of experience can be a bottleneck to more rapid progress and production during times of heavy work-load.

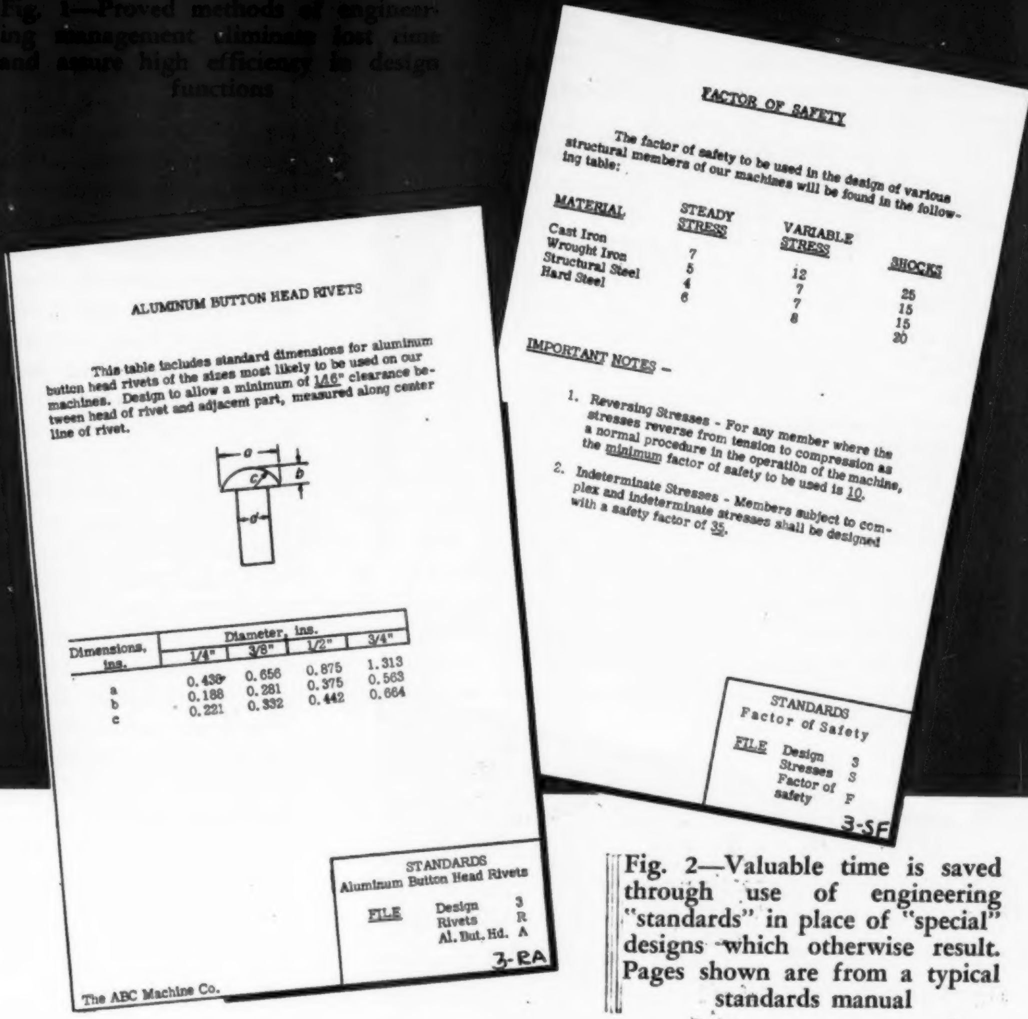
The solution to this particular problem, of course, is the preparation of what is usually known as a "Standards" manual. This book would contain all the answers to the questions a new man might ask. The manual can be something entirely new to the department or it can be a revision of one that had fallen into disuse because there had not been a particular need for it seen up to the present.

As an example of how an old book may be brought up-to-date, an excerpt from a report to a branch plant of a large manufacturing company, recommending certain inclusions in their manual, is pertinent:

1. Latest copy of the ASA-ASME, *Standard Drawings and Drafting Room Practice*
2. Drafting Practice text. The 3/1/39 issue is to be brought up-to-date, eliminating all parts that have been replaced by instructions from the main office, and rewriting the remainder
3. Up-to-date instructions on tolerances, allowances, finishes, etc.
4. Latest data on materials and heat treatments

Courtesy Boeing Airplane Co.

Fig. 1—Proved methods of engineering management eliminate lost time and assure high efficiency in design functions



FACTOR OF SAFETY

The factor of safety to be used in the design of various structural members of our machines will be found in the following table:

MATERIAL	STEADY STRESS	VARIABLE STRESS	SHOCKS
Cast Iron	7	12	25
Wrought Iron	5	7	15
Structural Steel	4	7	15
Hard Steel	6	8	20

IMPORTANT NOTES –

1. Reversing Stresses - For any member where the stresses reverse from tension to compression as a normal procedure in the operation of the machine, the minimum factor of safety to be used is 10.
2. Indeterminate Stresses - Members subject to complex and indeterminate stresses shall be designed with a safety factor of 35.

STANDARDS
Factor of Safety
FILE Design 3
Stresses S
Factor of safety F
3-SF

ALUMINUM BUTTON HEAD RIVETS

This table includes standard dimensions for aluminum button head rivets of the sizes most likely to be used on our machines. Design to allow a minimum of 1/16" clearance between head of rivet and adjacent part, measured along center line of rivet.

Dimensions, ins.	Diameter, ins. 1/4"	3/8"	1/2"	3/4"
a	0.438	0.656	0.875	1.313
b	0.188	0.281	0.375	0.563
c	0.221	0.332	0.442	0.664

STANDARDS
Aluminum Button Head Rivets
FILE Design 3
Rivets R
Al. But. Hd. A
3-RA

The ABC Machine Co.

Fig. 2—Valuable time is saved through use of engineering "standards" in place of "special" designs which otherwise result. Pages shown are from a typical standards manual

5. Editing of the engineering information in the old manual commonly found in engineering handbooks
6. This branch's proposal of standard instructions for keys and keyways
7. Latest information on the procedure for patents.

This manual will be contained in a proper type ring binder with tab and section divider sheets for easy insertion of new material and it will be of such type that the manual will lay flat while in use.

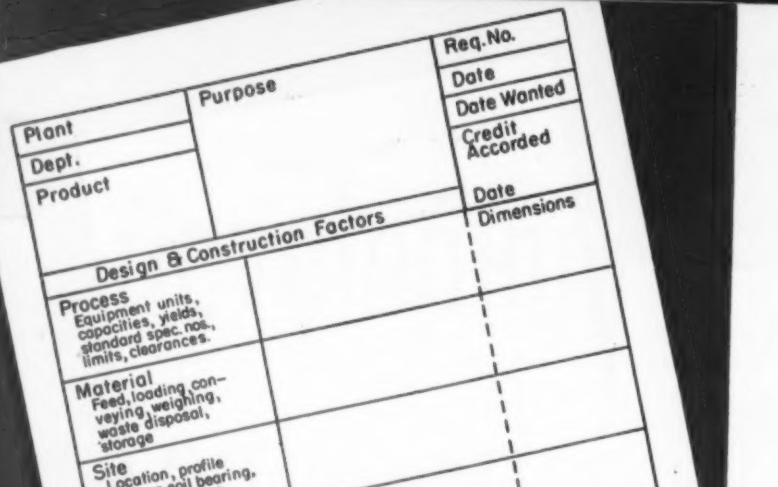
The material to be included in a manual of this type will vary according to the nature of the engineering department involved. In some cases repetitive details may be shown with a table of sizes covering the range of anticipated installations so that no time is wasted by many men continually designing similar details, *Fig.* 2.

The manual must of course be revised constantly to reflect improvements in design and design technique. This is the responsibility of management and just as old or outmoded machines must be replaced with new, so must improved design, methods and materials be sought out and utilized. Most changes will not be revolutionary; many engineers will agree that even the most radical innovation, in a practical industrial development, generally represents a single further step ahead in the state of the art.

In addition to the standards manual, the fund of information to be found in manufacturers catalogs, magazines, handbooks, and textbooks should be made readily available by means of an adequate engineering library. It may be concentrated in a central library or divided into several small individual libraries, depending on the particular requirements of the department.

With the manual at hand, the new man is off to a good start. He is able to proceed without asking too many questions and he develops a feeling that he is working for an alert organization that takes at least one important step to help him turn out better work faster.

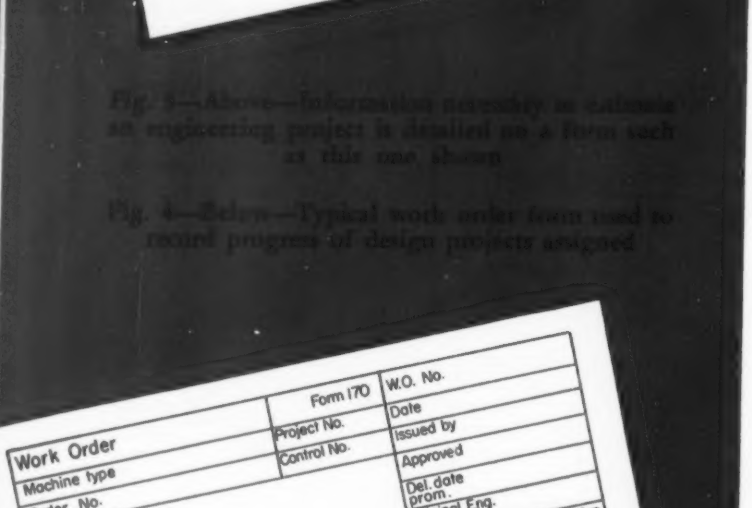
Streamlining Engineering Procedure: Once he has



Req. No.
Date
Date Wanted
Credit Accorded
Date
Dimensions
Plant
Dept.
Product
Purpose
Design & Construction Factors
Process — Equipment units, capacities, yields, standard spec. nos., limits, clearances.
Material — Feed, loading, conveying, weighing, waste disposal, storage
Site — Location, profile contour, soil bearing, exist. buildings, pipe connections.
Erection — Type of structure, load, walls, surplus materials available.
Services — Water, gas, steam, elec. current, dust collect., soil and waste, safety.
Signed
Alternatives
Request For Engineering Estimate
Form-E10

Fig. 3—Above—Information necessary to estimate an engineering project is detailed on a form such as this one shown

Fig. 4—Below—Typical work order form used to record progress of design projects assigned



Work Order
Form 170
W.O. No.
Machine type
Project No.
Date
Order No.
Control No.
Issued by
Approved
Customer or Engineering request by
Del. date prom.
Original Eng. promise
Present Eng. promise
Estimated hrs. to complete
Job Description
Draftsmen
Est. hours
Scheduled to start
Completed
Prints issued
E. O.
Prototype
Remarks

become established, the next problem is to use his time in the most efficient manner possible. Fortunately there is a certain body of techniques that has gradually evolved and is available to the modern-minded chief engineer who insists on getting his department up to the same standards of productivity by his colleagues in sales and manufacturing.

At the outset it must be said that these methods are simple to install and to operate. They assume that every man in the organization, from draftsman to project engineer, wants to make good on his job and with an understanding of what is being aimed at, will not only cooperate in applying more effective management methods, but will contribute constructively to their creation. The methods must be built up from the bottom, firmly, without disturbance, as a steady pace is often the key to long-range success.

The methods start at the beginning of a job. Often one of the most difficult tasks is to find out exactly what is to be designed. Most engineers are familiar with the time-consuming partial exchange of information between the design office and the customer or even between several elements of the same office. One solution, *Fig.* 3, is a form developed to meet the needs of a particular company, carrying all the information necessary to estimate the engineering work.

As soon as a job is to be definitely assigned to a design department, some record of its initiation and progress must be kept in the administrative office. A work order form, shown in *Fig.* 4, may be used for this purpose. This may show a complete job or only a part. The job may be divided according to the various divisions of the design office, e.g., structural, hydraulic, electrical, etc., and the order may be accompanied by various other papers giving a more detailed description of the work to be done. In any case, it provides on one sheet all the information to identify and keep track of the work.

The work order form contains considerable space for dates, both promises and actual dates of starting and completion. This concentration on time and planning is the backbone of the methods to increase production. Project scheduling, as set up in virtually every design office, is an element in such planning, but the usual project scheduling is far from providing the management strength that is given by the full conception of planning. When this planning and scheduling is used as a main instrument of management, in the sense that we are developing here, it means forecasting in a reasonable, studied way the use of time by persons responsible.

In general, whatever the local nomenclature or grouping of services, most or all of the following functions have to be dealt with:

- Preliminary investigations
- Studies of prior art
- Calculations and specifications
- Preliminary design
- Experimental models
- Final Design
- Detailing (drawings and B/M's)
- Prototypes of final models
- Process engineering
- Parts lists and technical manuals.

Time will have to be allotted for each of the functions required and a total estimated for the complete job.

Planning and Scheduling: It would be pleasant, although very unusual, if each new job was initiated just as the present one was being completed. More often they are introduced when most of the men are tied up in equally important work and the new work must be fitted in as expeditiously as possible. It is at this point that planning and scheduling become most essential.

It is important to emphasize that the time estimated for a job is a reasonable and responsible engineering estimate based on the assumption that each colleague will live up to his share of the responsibility. It is not a hopeful guess, nor is it an irresponsible assumption that everything will go wrong and that anticipated times should be padded for contingencies.

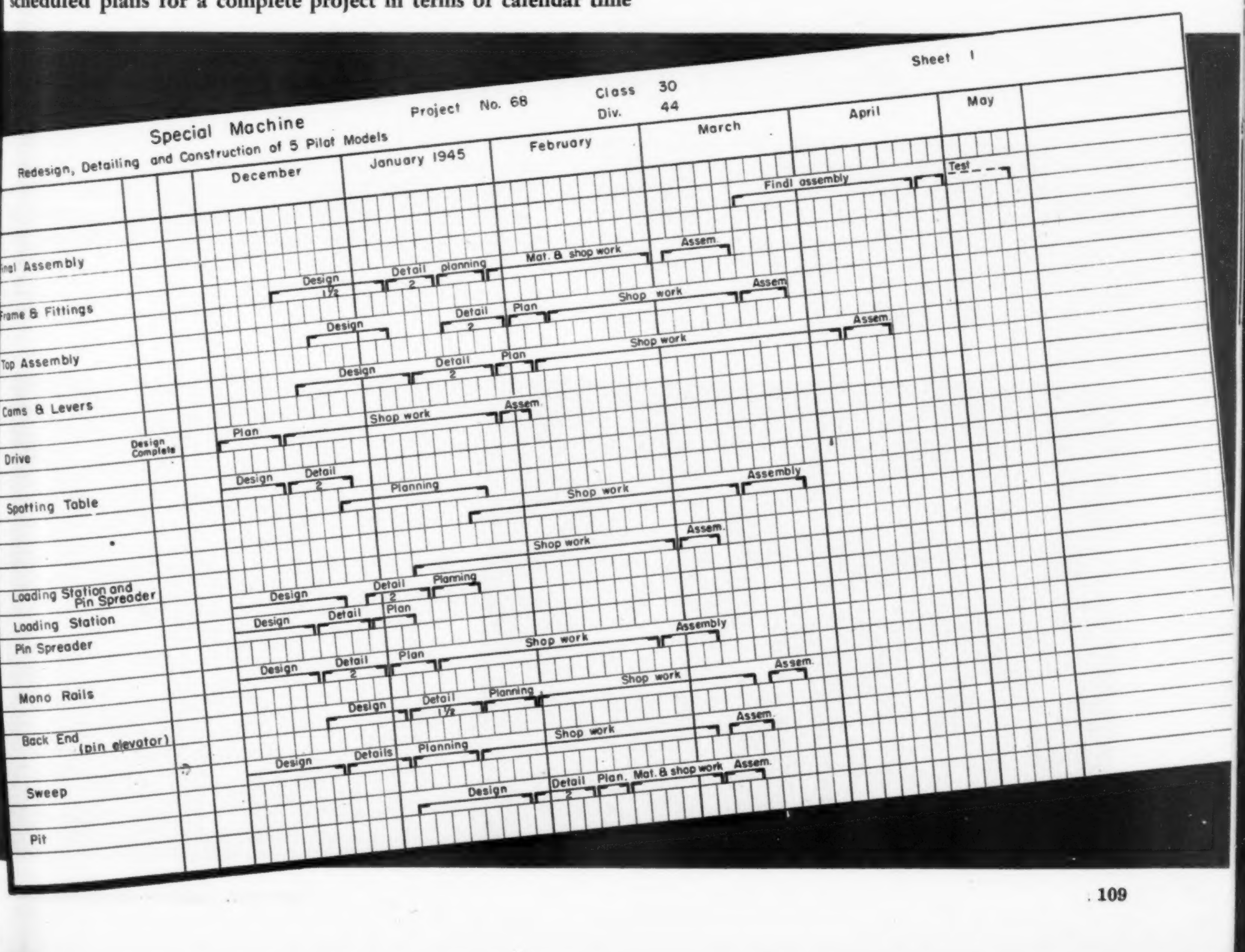
The lower-level planning of the work is the responsibility of those directly involved—the group leaders or squad bosses. An administrative assistant or scheduler may keep the formal schedule charts and take care of posting progress but the foresighted thinking that goes into schedules is the major and continuing concern of every group leader and squad boss in the department.

While some chief engineers prefer tabulations of figures on man-hours and dates, it is more general practice to express time plans on a Gantt chart or graph. This chart emphasizes schedules in terms of calendar time and, in the case of an overall plan for a complicated project, is especially helpful in showing clearly the overlapping time periods of the several phases of the job, *Fig.* 5.

In practice, the schedule and the work order are used together. When a group leader receives the work order, everything will generally be filled in except the schedule. After consulting his schedule and noting the work load facing him, the group leader can estimate the schedule date for starting on this job. He then follows through by figuring what men may most easily be assigned to the new job, how long each will have to spend on his particular function, and figuring also an estimated date for completion. This information is filled in on the work order and also posted on his schedule. A copy of the work order may then be sent to the person in charge of the administration of the engineering department for his records.

When work finally starts on this new job, the group leader knows just who is going to do what and how long it can reasonably be expected to take. If he sees that he is not going to meet his anticipated dates,

Fig. 5—Gantt chart showing typical method for recording scheduled plans for a complete project in terms of calendar time



he is able to take action to speed up progress. This may involve a consultation with an executive to decide a policy question or it may mean assigning additional men to help. In any case, the group leader cannot fail to become more management conscious and to assume greater responsibility for the completion of drawings.

Executives who take time to build up this understanding of planning among group leaders will be rewarded many times over by performances that would have previously been considered unobtainable. They will also verify the experience of the successful design leader who has observed that the management of any undertaking involves but two major parts: planning and direction. The more careful the planning, the less time will be subsequently taken up by direction.

Psychological and Organizational Benefits: Our new man is now a part of the engineering department. He has his manual for reference and by looking at the group leaders schedule, he knows exactly what has been planned for the future and just how his work fits in with the overall picture. The group leader has discussed the schedule with him so our man does not feel he is being pushed but feels that he is part of the team. He too has a certain responsibility to get his work out on time, as after all, wasn't he the one who agreed it should be done.

Complete planning of the use of time will sometimes also clear the way for major improvements in the organizational structure. For example, a department head may take over an organization divided into autonomous product groups, each fully staffed, and question whether he might obtain greater efficiency by centralizing certain functions.

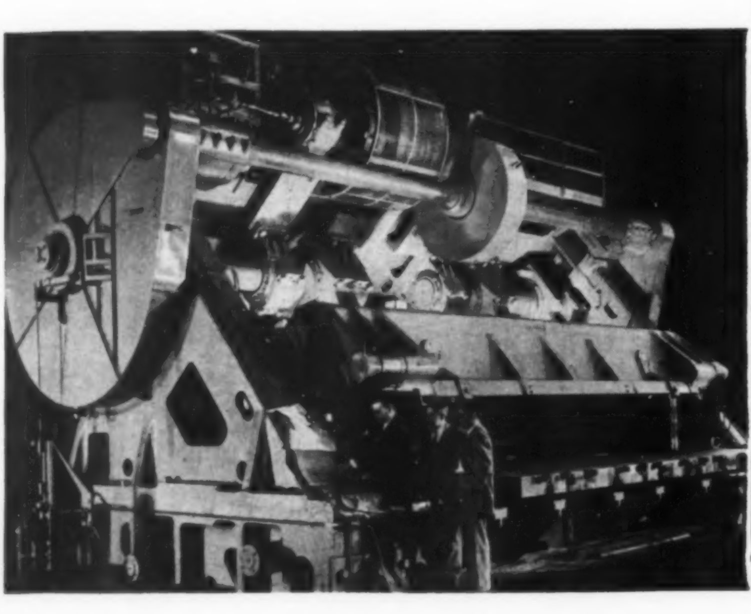
It is sometimes considered that greatest flexibility and usefulness of engineering personnel is obtained when all but the research staff and senior project engineers are regarded as a single personnel pool—engineers, draftsmen, detailers, checkers, etc., from which individuals are assigned from time to time to specific project staffs. But theory provides no real answer in a specific case. Only concrete knowledge of how the time of individuals is now being utilized will enable the department head to determine whether he should seek useful improvement through organization changes.

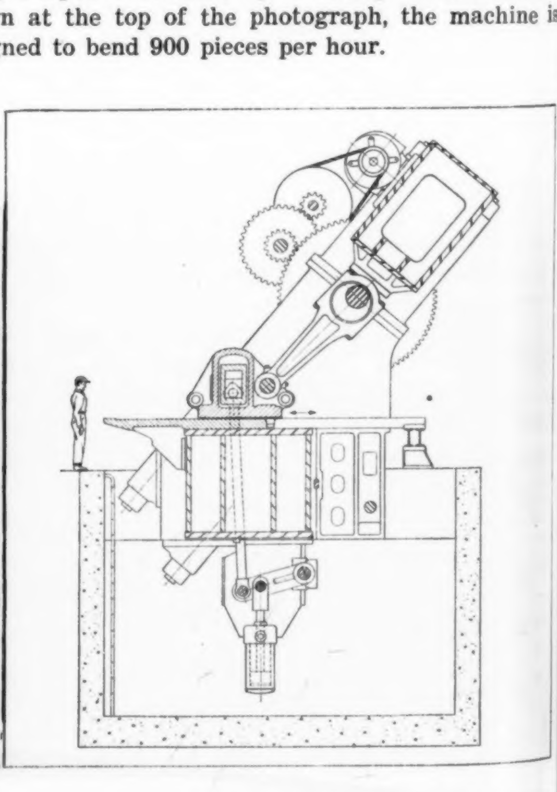
The introduction of the planning and scheduling methods and the preparation of the manual has a cumulative effect. In addition to the benefits accruing from these tangible changes, there is also an increase in production because of an increase of pride resulting from the recognition of the draftsman as an individual who is capable of taking a part, albeit a small one, in the planning of the project. He now has a definite goal, set by himself, against which to work and he is able to definitely prove his worth.

Reshapes Large Steel Sheets

USING a combination of mechanical, magnetic and hydraulic power, a "kickup" machine built by A. O. Smith Corp. reshapes steel sheets having a maximum length of 19 feet and from 5/64 to 5/16-inch thick. It can handle material up to 16 inches wide for truck and automobile frames, bending these long, narrow sheets edgewise.

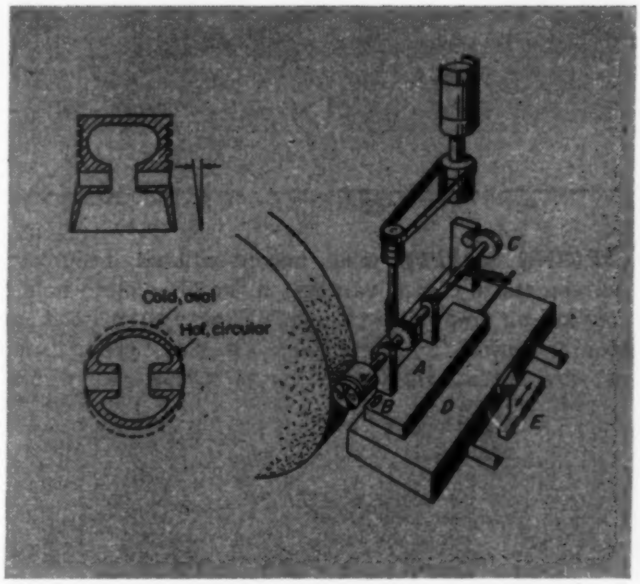
A 15-foot, 8-inch reinforced concrete pit contains much of the hydraulic mechanism which lifts the ram and assists in holding it down tightly on the piece being shaped. Powered by a 150-hp electric motor, shown at the top of the photograph, the machine is designed to bend 900 pieces per hour.





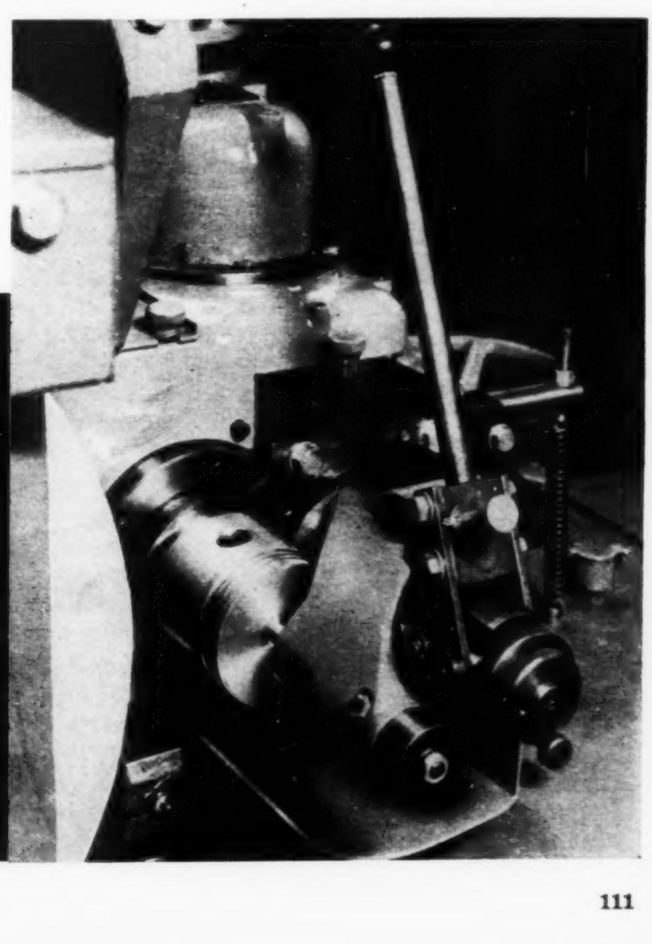
SCANNING the Field for IDEAS

CORRECT CYLINDRICAL SHAPE of a piston for an internal-combustion engine when working in the hot cylinder requires it to be ground with a slight taper from the open end, which is relatively cool, to the head, which is exposed to the high temperature of combustion. This taper compensates for the difference in expansion caused by the temperature gradient. Also, the influence of temperature on the masses of metal forming the bosses for the wrist-pin bearings causes these two sides to expand more than the thin walls of the other sides, necessitating an oval shape when cold.

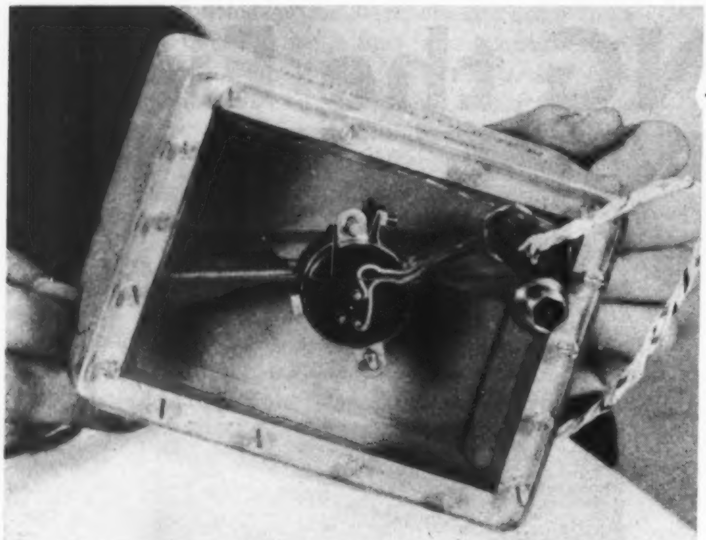


One machine for producing these corrections is shown below together with a diagrammatic sketch of the principle of operation as well as sectional views of a piston with these corrections. Designed by Arthur Scrivener Ltd., Birmingham, England, the grinder rotates the piston about a horizontal axis and at the same time oscillates it in a horizontal plane against the grinding wheel.

Referring to the diagram, the work is held in a fixture on a spindle assembly *A* which is pivoted at point *B*. Oscillating movement is obtained from cam *C* which is spring held against a fixed roller. Amount of pivoting determines the taper on the skirt and the lobes on the cam control the ovality of the walls. Cam *E* controls the position of the fixture slide base *D* for grinding and loading.

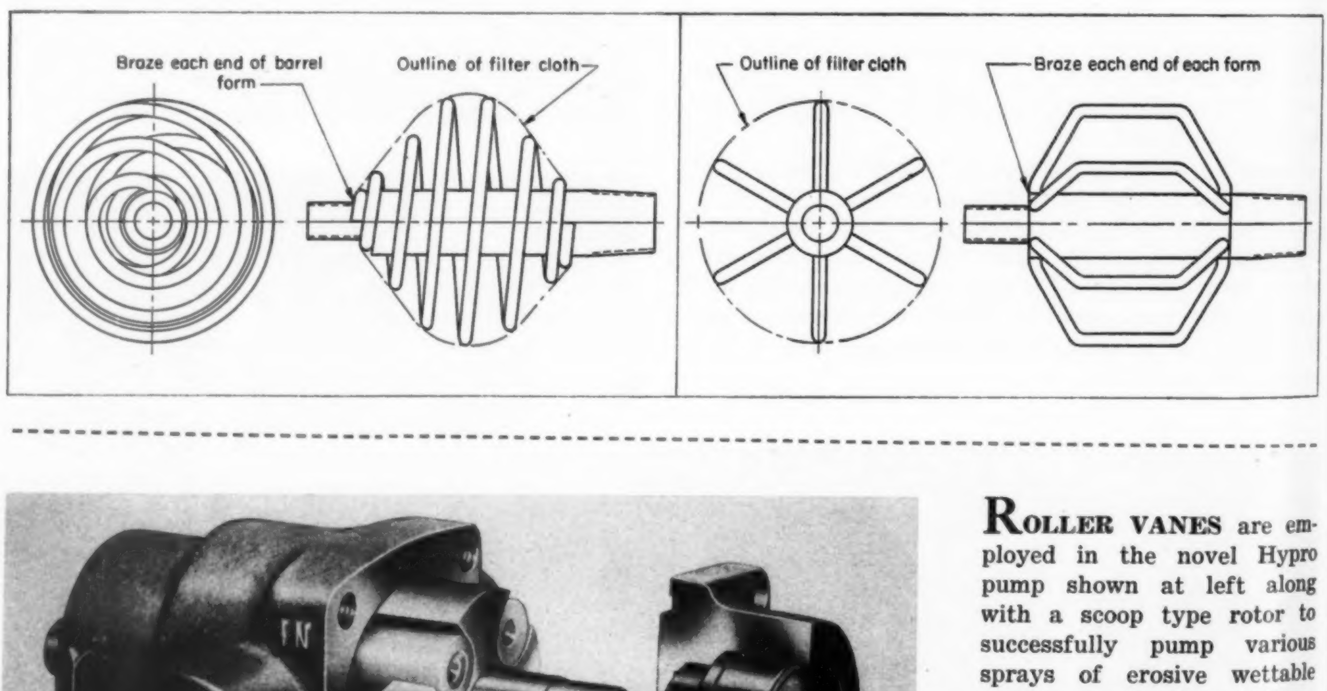


IDEAS



MINUTE ANGLES of deviation are precisely measured by the "horizontometer" shown at left. Developed by Morton Stimler of the Naval Ordnance Laboratory, the device is based on the property of a free fluid surface to remain horizontal. Two opposed floats attached to the shaft of a sealed multirevolution potentiometer operate partially immersed in the fluid. Suitable for measuring deviations of as much as 1800 degrees or, with two units at right angles, changes in pitch and roll, the design provides torque from restoring buoyant forces much higher than those developed with pendulum types for greatly increased sensitivity.

SPRING SHAPES meet unconventional design requirements to cost advantage in the filter cloth holder for an air regulator shown below. Designed by Hunter Spring Co. to replace a complex brazed group of wire forms, right, the barrel spring design provides more continuous support and is self positioning on the center tube, eliminating the need for fixtures in brazing. Owing to the fact that the barrel spring is produced on an automatic spring coiler, cost was reduced 60 per cent.

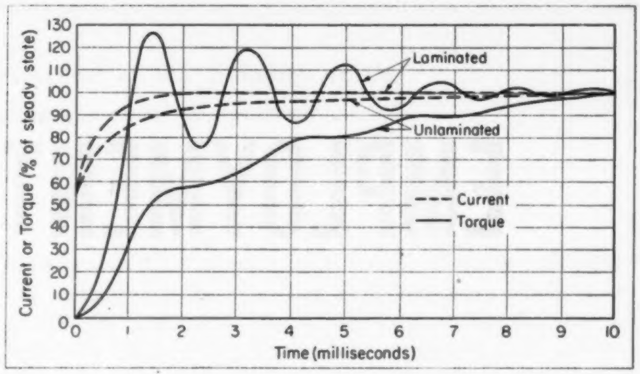


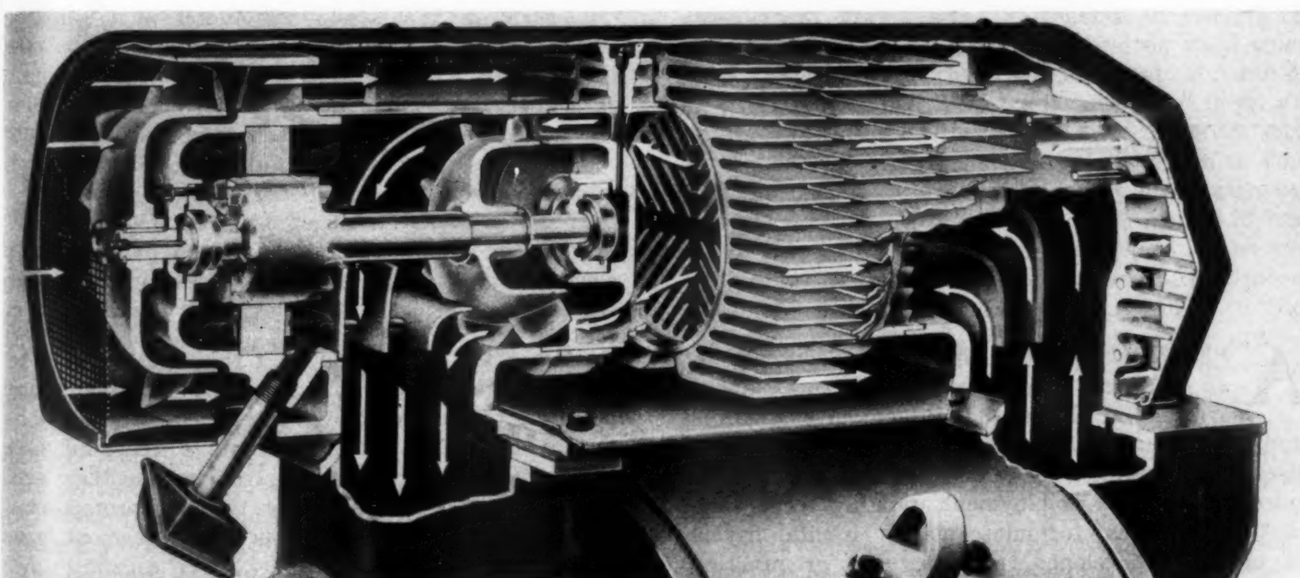
ROLLER VANES are employed in the novel Hypro pump shown at left along with a scoop type rotor to successfully pump various sprays of erosive wettable powder or abrasive solutions. Highly abrasion resistant nylon or Hycar rollers utilized require little lubrication, water being sufficient. With the rotor held against lateral movement by the bearing arrangement, wear is minimized and efficiency is high in the recommended range of pressures from zero to 200 psi.

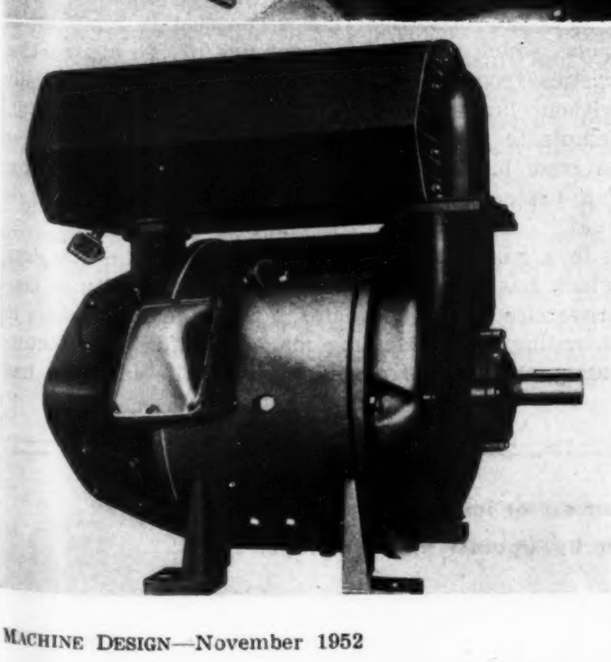
FASTEST RESPONSE from magnetic - particle clutches is obtainable by employing a laminated magnetic circuit. Recent studies by W. A. Notz of the National Bureau of Standards indicate that response time can be reduced to the order of 1.5 milliseconds for 63 per cent of maximum torque.

With other factors constant, torque developed by a magnetic-particle clutch is approximately proportional to the magnetic flux which, in turn, is roughly proportional to the exciting current. Comparative characteristics of laminated versus nonlaminated constructions are shown in the chart, right. Response time of the laminated design was found to be limited only by mechanical considerations, but that of the nonlaminated one was limited by eddy currents—because of these magnetizing current and output torque are slow in building up.

Assuming ideal mechanical resistance and stiffness in the output system, response speed can be maximized by maximizing the torque-to-inertia ratio. The analysis showed that for fastest response the inertia of the clutch disk should be three times the load inertia and that the magnetic circuit could be driven to saturation most rapidly by utilizing a laminated circuit.





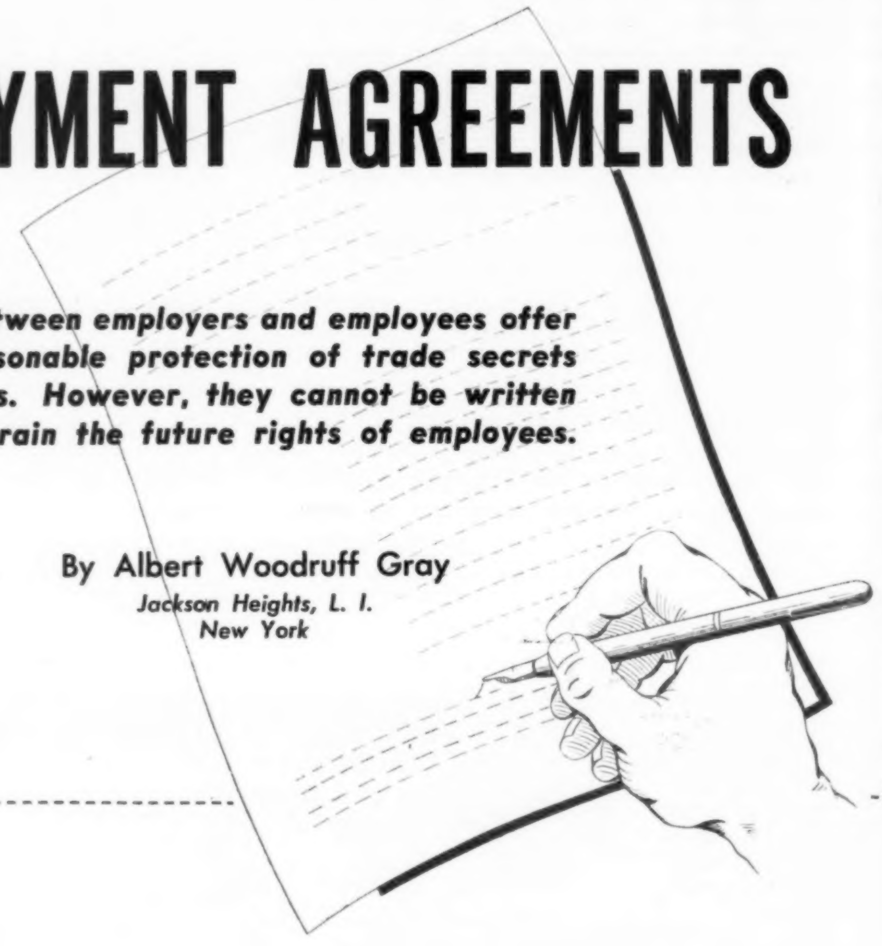


DUAL-COOLED DESIGN permits the manufacture of totally enclosed motors up to 150 hp—double the limits previously imposed. By building-in an integral heat exchanger as part of the motor, above, Reliance Electric & Engineering Co. has obtained increased cooling efficiency, permitting smaller motors for a given horsepower.

Air within the motor is circulated through the exchanger and over the motor coils at 2900 fpm. External cooling air is drawn in and pumped over the exchanger fins at 3300 fpm. A 1½-hp, 3-phase, 60-cycle motor drives the internal and external blower fans at 3600 rpm independently of the main motor.

Wide exposed aluminum fins utilized in the exchanger design along with the high velocity of air passage tends to keep the cooling surfaces free from fouling in service. Blade airfoil and fin structure have been designed to minimize vibration and noise.

EMPLOYMENT AGREEMENTS



Contracts between employers and employees offer fair and reasonable protection of trade secrets and processes. However, they cannot be written so as to restrain the future rights of employees.

By Albert Woodruff Gray
Jackson Heights, L. I.
New York

A PROVISION in the employment contracts of an electrical equipment manufacturer during World War II was that the employee, "Agrees that he will not at any time during said employment disclose to any one any information that he may acquire during said employment, relating to any one of the processes, formulas, plans, circuits, devices or methods, developed, manufactured or practiced at any time by said corporation in its business and he will not use any of said processes, formulas, plans, circuits, devices or methods or his knowledge of the same except in the course of his employment by the corporation."

A foreman who had learned many of the manufacturing secrets of this company resigned. Later he entered the employ of a competitor. The first employer sued to prevent both the former employee and the competitor from disclosing the information that the employee had stipulated by contract not to divulge.

This contract the Federal court refused to enforce, stating, "It is quite clear that the contract goes beyond the protection of trade secrets and embraces anything that the employee saw or learned during his employment. The agreement puts a restraint upon the employees' right to labor or exercise their skill greater than is necessary for the fair protection of the employer and therefore such an agreement is unenforceable by injunction.

"The law is settled that a contract in restraint of labor, which seeks to prevent one of the contracting parties from exercising his skill or labor generally without limitation as to place or time, or which attempts to put a restraint upon his right to labor or exercise his skill greater than is necessary for the fair protection of the other party to the contract, is void."

In a case decided long ago in Massachusetts, and which has often since served as an authority in controversies of this character, a man had spent years in inventing and perfecting machinery for manufacturing gunny cloth from jute butts. The inventor had

" . . . persons under a contract, either express or implied . . . cannot gain knowledge of the secret and then set it up against their employer . ."

". . . restraint must be for the protection of the property of the employer . . . and not merely the suppression of competition . . ."

constructed a large factory, equipped it with these machines and employed a foreman under a contract in which the foreman agreed that he would give no information to any person, directly or indirectly, regarding the machinery or its operation and "consider all of said machinery as a secret to be used only for the benefit of the employer or his assigns, and by all means within his power prevent other persons from obtaining any information in regard to it such as would enable them to use it."

Later, the foreman left this employer to accept a position with a competitor who was planning the construction of another factory equipped with machines of this type constructed according to plans and drawings the foreman had secured. Suit was brought by the former employer for an injunction against this disclosure of secrets.

"There is no doubt whatever," said the Massachusetts court, granting an injunction against these disclosures, "that when a party having a trade secret employs persons under a contract, either express or implied, those persons cannot gain knowledge of the secret and then set it up against their employer.

"Whatever may be the limit or effect of this employee's obligation to serve, he is bound by his contract never to disclose the secret confidentially imparted to him during the term of his actual service and this part of the agreement may be specifically enforced, even if the other part is not."

The following year another contract of this same character was again before the courts of that state. This agreement had been made by the owner of patents on twist drills and collets under which the patentee contracted to transfer to a manufacturing company these patents and his rights to any letters patent for any drills or improvements.

The company in return agreed to pay the inventor $5000 in thirty days, a further payment of the same amount out of the earnings of the company and an additional $4500 in three annual installments. The inventor continued in the employ of this company for three years, after which he resigned and set up the manufacture of these drills in another state.

In its decision of the action brought to prohibit the continued violation by this inventor of his agreement, the Massachusetts court, following the authority of the previous case, said of this contract, "It was not in restraint of trade nor contrary to public policy that this employee should contract for his exclusive services in this respect. He could not have obtained the consideration paid him if it had been understood that this contract which he has violated, had no validity. He is appropriating to himself a part of that which he has sold to his employers and which is valuable property to them."

The criterion by which is determined the validity of an agreement not to accept employment with a competitor, is whether or not the restraint imposed on the employee is greater in its duration or in the territory affected than adequate protection requires. Further, the restraint must be for the protection of the property of the employer, as trade secrets, good will or other incidents of the business and not merely the suppression of competition.

In a famous decision in the Federal courts William H. Taft, then a Federal judge, said of the validity of employment agreements, "Covenants in partial restraint of trade are generally upheld as valid when they are agreements by . . . an assistant, servant or agent not to compete with his master or employer after the expiration of his time of service. Before such agreements are upheld however, the court must find that the restraints attempted thereby are reasonably necessary . . . for protection from the danger of loss to the employer's business caused by the unjust use on the part of the employee of the confidential information acquired in such business.

"We do not see how a better test can be applied to the question whether this is or is not a reasonable restraint of trade than by considering whether the restraint is such only as to afford a fair protection to the interests of the party in favor of whom it is given and not so large as to interfere with the interests of the public.

"Whatever restraint is larger than the necessary protection of the party requires can be of no benefit to either. It can only be oppressive. Whatever is injurious to the interests of the public is void on the ground of public policy.

"This very statement of the rule implies that the contract must be one in which there is a main purpose to which the covenant in restraint of trade is merely ancillary. The covenant is inserted only to protect the parties from the injury which in the execution of the contract or enjoyment of its fruits, he may suffer from the unrestrained competition of the other."

An employee of a New Jersey manufacturer had

". . . persons who induce the employee to disclose the secret . . . will be enjoined from making further use of the information . . ."

". . . if the contract is founded on valid consideration and a reasonable ground of benefit . . . it is free from objection and may be enforced . . ."

agreed with his employer not to reveal secret manufacturing methods. Nine years later this employee became dissatisfied, resigned, and two weeks later accepted employment with a competing firm.

In granting the first employer an injunction against the disclosure by the former employee of these secrets and against the competing firm from using or divulging any information derived from this employee, the New Jersey court said, "The right of the manufacturer, whose goods are made by an unpatented secret process, to protection by injunction against the divulging of his secret in a proper case, is now established by a well considered line of cases in England and in several states.

"These cases establish the principle that employees of one having a trade secret, who are under an express contract or a contract implied from their confidential relation to their employer not to disclose that secret, will be enjoined from divulging the same to the injury of their employer, whether before or after they have left his employ; and that other persons who induce the employee to disclose the secret, or that knowing his disclosure is in violation of the confidence reposed in him by his employer, will be enjoined from making any use of the information so obtained although they might have reached the same result independently by their own experiments or efforts."

The agreement under which this employee was forbidden to disclose the methods and trade secrets of his employer was well within the rule laid down in this famous decision by Taft.

In contrast to this is the aforementioned employment contract of the electric equipment manufacturer in which the employee agreed not to disclose any information he might receive irrespective of its source. That contract, which the Federal court refused to enforce, had projected its "thou shalt not" beyond the limits necessary for the protection of the interests of the employer.

In a recent Wyoming action for an injunction against a violation by a former employee of a contract of this character the court in that state said, "It has been said that sound policy encourages employees to seek better jobs from other employers or to go into business for themselves. Contracts which hinder their so doing are strictly considered and rigidly scanned and are declared void unless necessary for the reasonable protection of the employer." This ruling supplemented a statement of law principle made nearly a hundred years ago by the United States Supreme Court.

To illustrate this principle, two business partners are engaged in manufacture of a certain article by secret process, and they agree to separate. One of the terms of their separation is that one of the parties shall not sell the article in Massachusetts where the other resides and carries on business and that the other shall not sell the article in New York where his former associate is to reside and carry on business.

Can there be any doubt that such an agreement would be valid and binding? Possibly so, for there are two principal grounds on which it may be considered a contract in restraint of trade, and, therefore, is void as against public policy. One is injury to the public by being deprived of the restricted party's industry; the other is the injury to the party himself by being precluded from pursuing his occupation and thus being prevented from supporting himself and his family. But if neither of these evils ensue and if the contract is founded on a valid consideration and a reasonable ground of benefit to the other party, it is free from objection and may be enforced.

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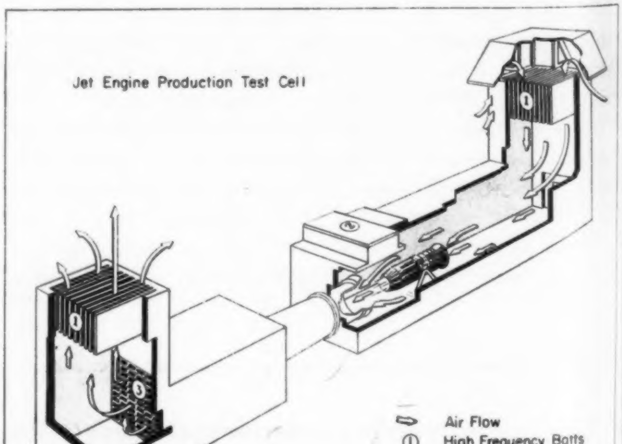
7. Oregon Steam Navigation Co. v. Windson, 87 U.S. 64

Turbojet Test Cell

A SYSTEM of blocks and grid-like "batts" lowers the sound level of turbojet engines under test. The cells at G-E's Lockland, Ohio, jet center are reinforced concrete made from a special aluminous cement capable of withstanding exhaust blasts which send cell exhaust temperatures up to 1000 F in 8 seconds. Air enters the engine at 100 mph and exhausts at 1300 to 2000 mph. From the engine the exhaust gases are directed to a cylinder "augmenter" which carries them to the exhaust stack.



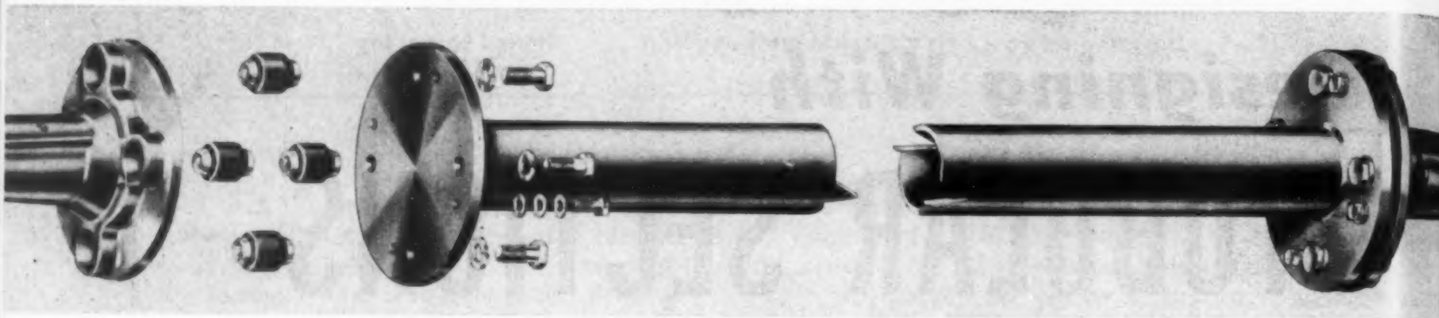
Designing With
TUBULAR SECTIONS

New and unusual applications emphasize the many design opportunities of this basic material form

By Leo F. Spector
Assistant Editor
Machine Design

Fig. 1—The boom of this giant dragline built by Stewart and Lloyds Ltd., 282 feet in length, is an all-welded triangular cantilever jib of high-tensile steel tubing. Main compression and tension members are seamless cold-drawn, normalized tubes; subsidiary bracing and support members are seamless, hot-finished tubes; and lattice bracing members are welded tubes; in sizes from $1\frac{1}{4}$ to 16 inches. Typical tubular joint member used is shown in close-up view

IN REACHING today's goals of greater simplicity and economy in design, success may often depend on recognition of the opportunities offered by tubular shapes. Highlighted by some of the ever-increasing applications both as static and dynamic machine components, *Fig.* 1, the versatile character of mechanical tubing as a basic engineering material attains full significance. By continuing appraisal of the inherent properties of the tubular shape in conjunction with the supplementary functional and producibility possibilities afforded by design ingenuity, efficient utilization of available tubular materials in



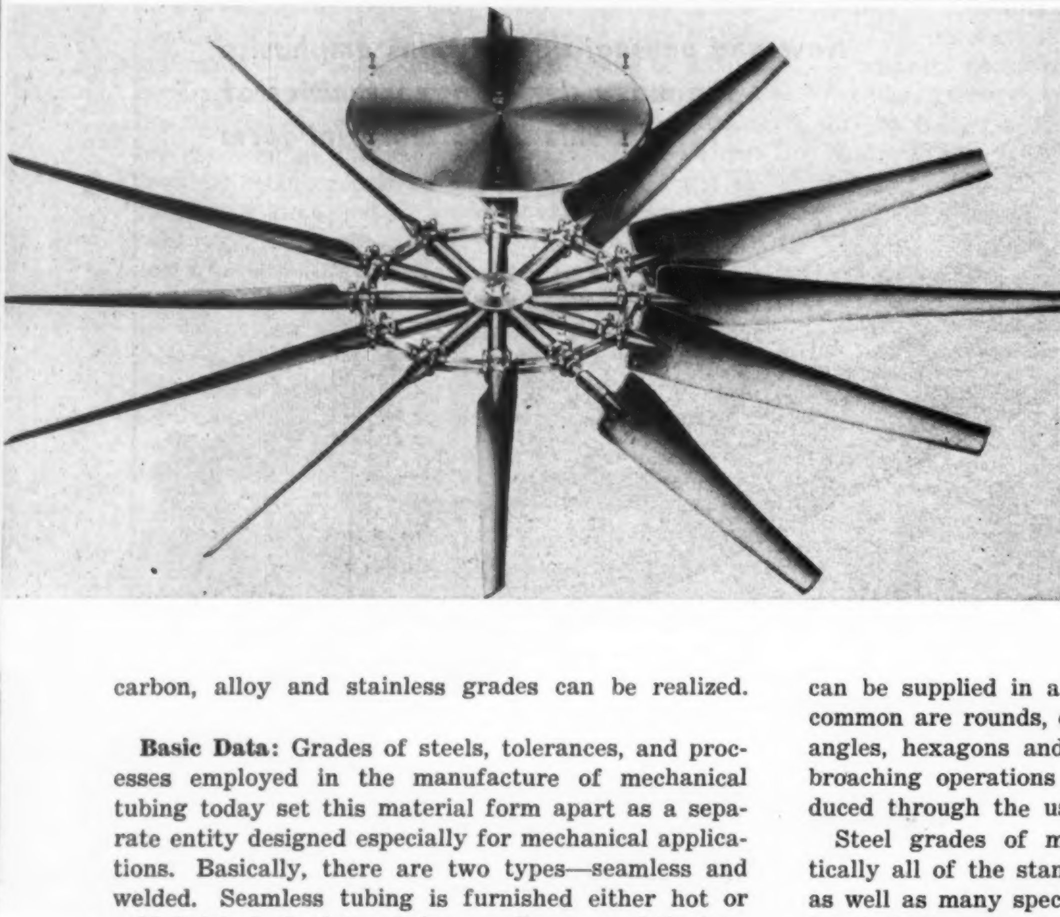


Fig. 2—Above—High torsional capacity, lightweight and vibration characteristics make tubing ideal for power transmission members such as this long drive shaft used on induced-draft cooling towers

Fig. 3 — Left — Tubular spokes in the welded hub of this 72-inch, 12-blade fan absorb complex bending loads

carbon, alloy and stainless grades can be realized.

Basic Data: Grades of steels, tolerances, and processes employed in the manufacture of mechanical tubing today set this material form apart as a separate entity designed especially for mechanical applications. Basically, there are two types—seamless and welded. Seamless tubing is furnished either hot or cold finished, the latter being a refinement of the hot-rolled tube through cold drawing and featuring better finish and closer dimensional accuracy. Welded tubing is furnished either hot or cold rolled, the name in each case signifying the class of steel sheet from which the tube is formed. Here again the cold-rolled tube has improved tolerances, finish and properties. Further refinements in finish and dimensional tolerances can be obtained by operations such as cold sizing and grinding.

Available mechanical tubing encompasses a wide variety of sizes, shapes, analyses, heat treatments, dimensional accuracies, and surface finishes. The general range of standard sizes for welded tubing (round) is 3/8-inch OD by 0.028-inch wall to 5-inch OD by 0.259-inch wall; for seamless tubing (round), 3/8-inch OD by 0.035-inch wall to 10½-inch OD by 1-inch wall. Special sizes extend these ranges to tubes as small as 0.008 inches OD with a 0.002-inch wall and as large as 35-inches OD with a 4½-inch wall. Cross-sectional shapes in the standard ranges can be supplied in a wide variety; among the more common are rounds, ovals, streamlines, squares, rectangles, hexagons and octagons. Planing, shaping or broaching operations may often be minimized or reduced through the use of these special shapes.

Steel grades of mechanical tubing include practically all of the standard AISI commercial analyses, as well as many special analyses from plain low carbon to high carbon, alloy and stainless. Usually supplied on the basis of end requirements with compositions and heat treatments being determined by the individual steel manufacturers, tubing analyses and tempers vary widely to meet mechanical design requirements. Because of this wide variety of requirements, standard specifications have not been developed for these products.

Functional Properties: As a machine or structural component, mechanical tubing offers fundamental advantages in:

1. Torsional strength.
2. Bending strength.
3. Dynamic loading.
4. Material distribution.
5. Impact loading.
6. Multipurpose application.

TORSIONAL STRENGTH: Compare dto other structural sections of the same material and weight, round tubular sections have a considerably greater torsional

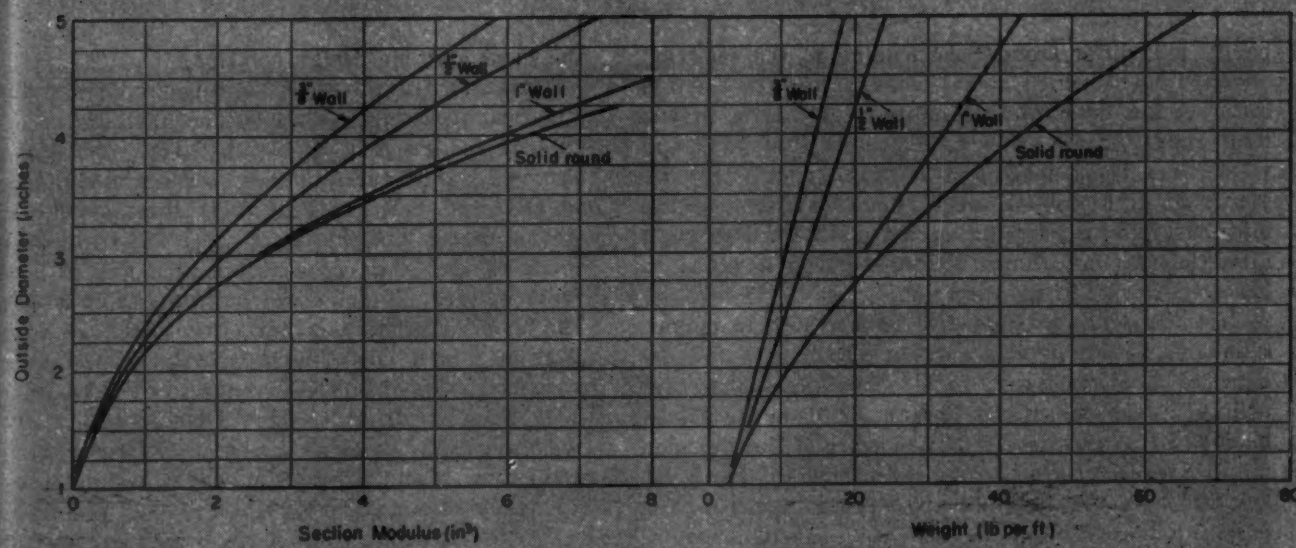
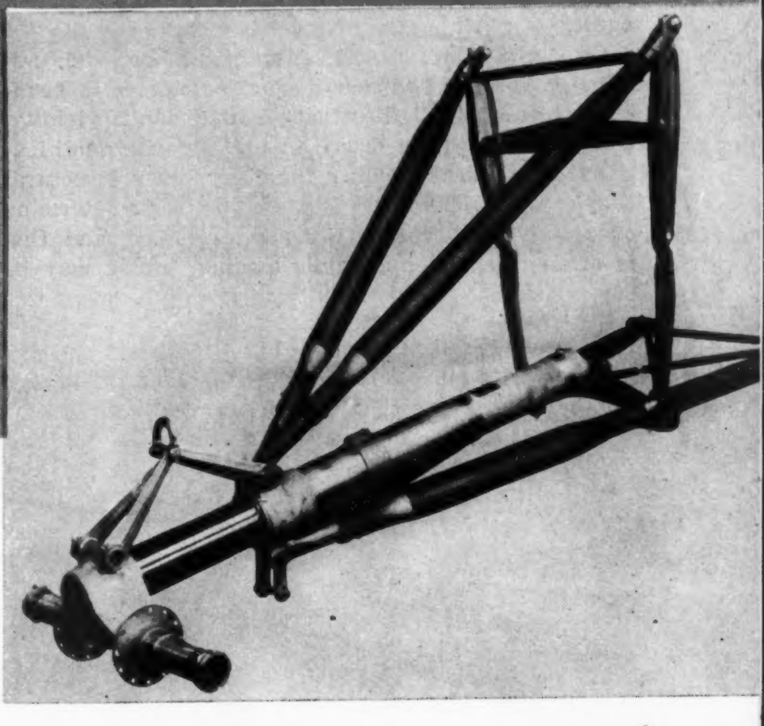


Fig. 4—Above—Curves comparing weights and section moduli for different round tube wall thicknesses and OD with those for solid rounds show the superior weight-strength characteristics of the tubular section

Fig. 5—Right—The four swaged-down tubular struts in this Fairchild C-119B landing gear assembly built by Cleveland Pneumatic Tool Co. serve to strengthen and position the installation



capacity. This superiority derives from the fact that torsional loads are carried primarily by the outer portion of the loaded member, the inner portion adding little to the load capacity but greatly to the weight. Also, when under dynamic loading, as in a drive shaft, tubing has a higher frequency and smaller amplitude of vibration than any other section, including a solid round. The combination of these two properties makes the tube ideal for power transmission applications.

Powering of large horizontal cooling tower fans on induced draft towers is accomplished by long drive shafts which extend outside the fan diameter to the power unit. Since intermediate support bearings required with solid-steel shafting would be impractical, large-diameter thin-wall tubing is utilized. This has proved satisfactory not only from weight but also life and vibration standpoints. A typical shaft with drive yokes is shown in *Fig.* 2.

Torque capacity of thick-walled tubular drive shafts, derived from the basic torsional equation, is

$$T = \frac{\pi s (D^4 - d^4)}{16 D} = 0.196 s \frac{D^4 - d^4}{D}$$

where T = torque capacity, pound-inches; s = allowable shear stress, psi; D = outside diameter, inches; and d = inside diameter, inches. Selection of s will depend on the material and the desired factor of safety. Caution should be exercised in applying this equation to thin-walled tubes which may fail in buckling below the maximum shear limit. Critical speeds of tubular shafts may be computed from

$$N = \frac{4.705(10^6)}{L^2} \sqrt{D^2 + d^2}$$

where N = critical speed, rpm; L = length, inches; D = outside diameter, inches; and d = inside diameter, inches. For safe speeds the results obtained from this formula should be reduced a minimum of 15 per cent.

BENDING STRENGTH: As a beam, tubular members have the least weight for a given strength and stiffness when there is to be equal loading in any direction. This property also is attributable to the fact that the outer fibers of the loaded member carry the bending load while the inner portion does little but add weight. Considerations involved in the application of a tubular beam with its uniform bending ca-

pacity will differ from those leading to the selection of an I-beam which has been designed to have a high section modulus and moment of inertia in one plane only. This is also true for the rectangular beam which has been designed for loading in only two planes at right angles. A further advantage of the uniformity of the tubular section is obtained through its efficiency in column action—equal resistance to bending stresses in any direction.

Complex bending loads, fluctuating both in magnitude and direction pose another problem in the design of cooling tower fans. Tubular spokes are utilized in the hub of the 72-inch, 12-blade fan shown in *Fig.* 3, to provide an efficient torque-transmission unit. A steel casting provides a large center weld-attachment for the spokes and welded braces tie the outer ends together. The fan blade shanks are inserted in the split tube ends and U-bolt clamps are drawn tight to secure the blade at the desired pitch angle.

DYNAMIC LOADING: Because of its material uniformity, torsional capacity, equal resistance to bending stresses in all directions, and dynamic rigidity, round tubing comes closer to meeting the qualifications of an ideal member than any other structural shape. When it is difficult or impossible to determine accurately the types of imposed stresses, and this is often true under dynamic loading, tubing may be the most suitable design solution.

MATERIAL DISTRIBUTION: A better understanding of the weight-strength distribution advantages of the tubular section may be obtained from the graphs of *Fig.* 4. For circular shapes, section moduli are indicative of both bending and torsional strengths (polar moment of inertia is twice the moment of inertia of the beam cross section). Thus, it may be seen that for a 3-inch solid round weighing 24 pounds, the same approximate strength could be obtained with a 3-inch OD, 1-inch wall tube weighing 21.4 pounds; a 3¼-inch OD, ½-inch wall tube weighing 14.7 pounds; or a 3½-inch OD, ⅜-inch wall tube weighing 12½ pounds—almost a 50 per cent weight reduction. Comparisons for equal weight can also be easily obtained.

IMPACT LOADING: Under abnormal impact, tubular structures tend to localize and absorb the shock and prevent undue extension of damage throughout the structure. The continuity of cross section provides excellent energy absorption characteristics, minimizing the severe stress-raising effects which accompany shock loads. Evidence of this stability is the application of tubular struts to the landing gear

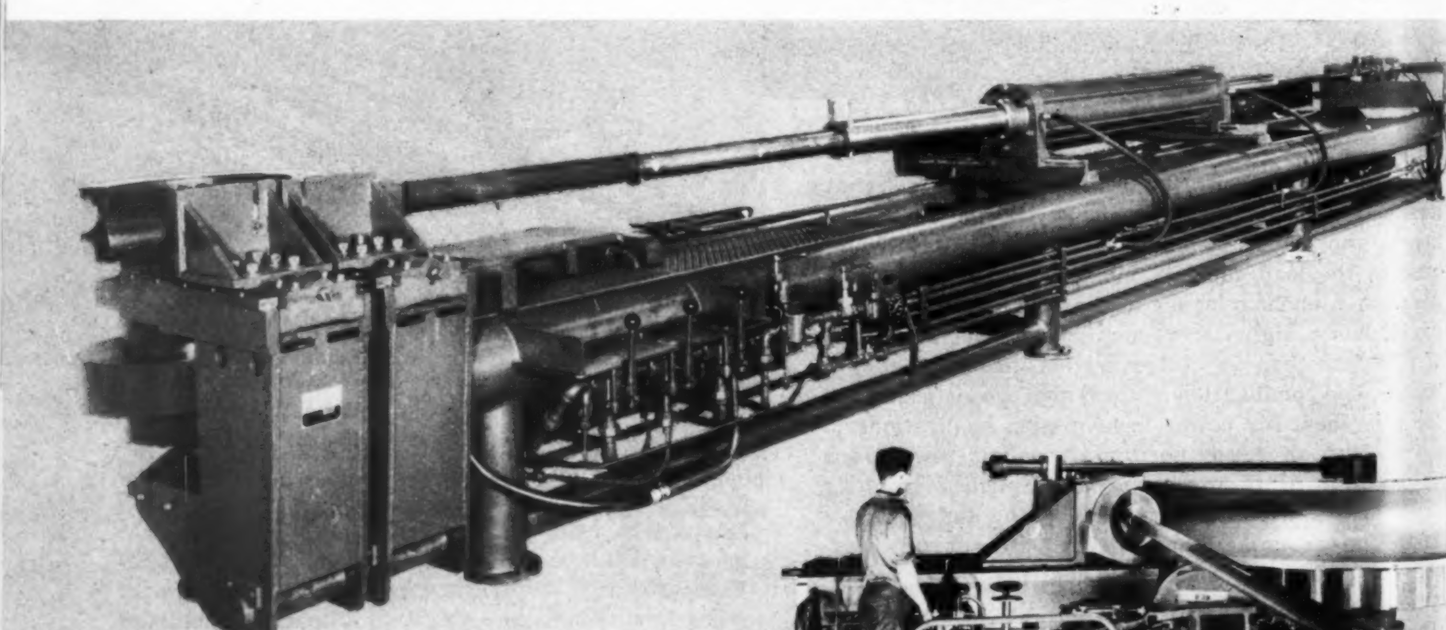
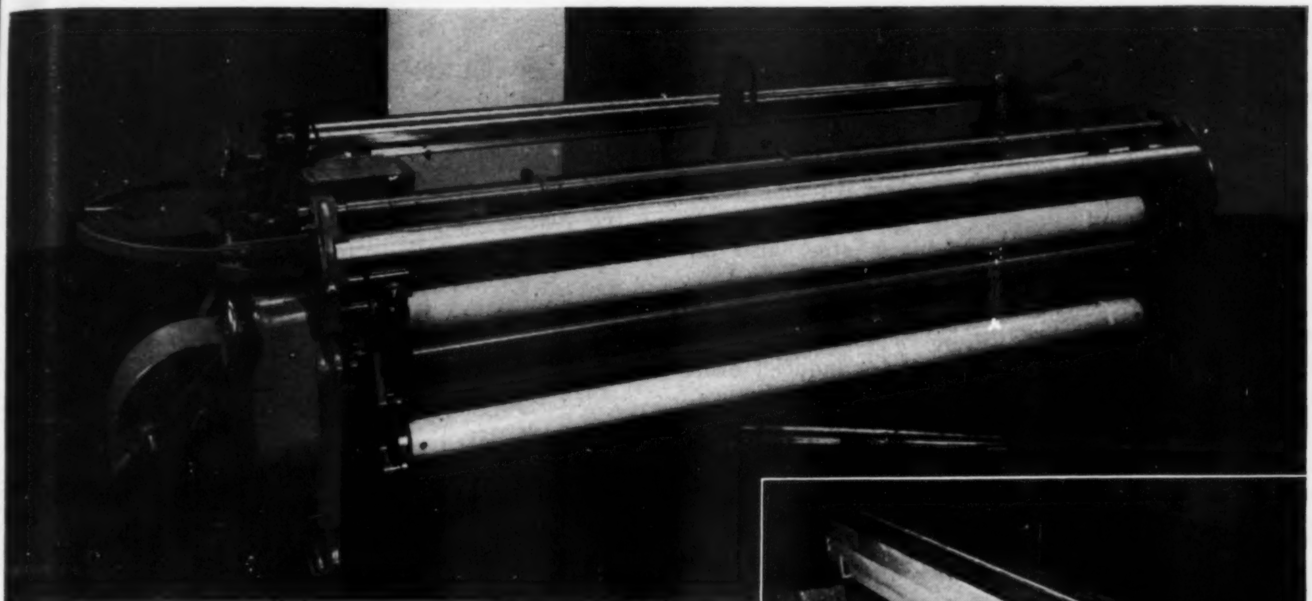


Fig. 6—Three different capacity hydraulic benders which utilize a tubular frame serving also as an oil reservoir. Machine at lower right develops a torque of 20,000,000 inch-pounds and bends 17-inch pipe



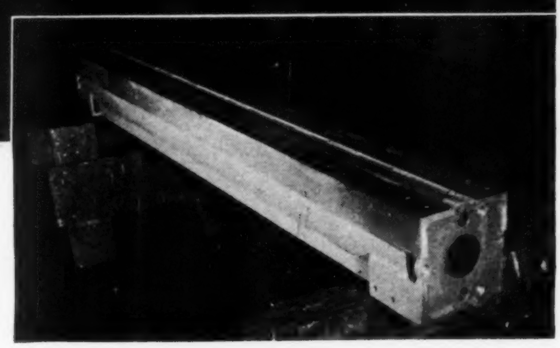


Fig. 7—Weaving machine employs tubing for rolls and also, with reinforcing angles and plates, for the main bed shown in the inset photo

of the Fairchild CB-119B airplane, *Fig.* 5. The four smaller diameter tubes, swaged down at the ends, strengthen and maintain position of the gear during landing. By using tubular components, maximum strength with minimum weight is obtained. Two of the tubes, as supplied, are 3.912-inch OD with a 0.179-inch wall, the other two are 4.956-inch OD with a 0.299-inch wall. Final finishing is accomplished by centerless grinding.

MULTIPURPOSE APPLICATION: Further evidence of the diversified utility of the tubular section is its adaptability as a multipurpose component. In addition to fulfilling the purposes of a primary load carrying member, it may at the same time be used to conduct or store liquids and gases, to carry electrical wiring, etc. The hydraulic bending machines in *Fig.* 6 employ this design flexibility to great advantage. Tubular members provide needed structural rigidity and also serve as oil reservoirs. The large surface area provides maximum radiating service for cooling the oil. Compared to previous machines made up of castings and structural sections, these redesigned units have greatly reduced weight and deflection, simplified assembly, improved appearance, increased working space, and above all, lowered costs.

An interesting sidelight in conjunction with bent tubing is the result of a series of tests conducted to determine the effect of cold-bending. It is well known that when tubing is bent the outer wall (tension) reduces in thickness while the inner wall (compression) increases. Stretching of the wall at the bend would seem to indicate a weakening at that point. However, hydrostatic tests carried to rupture, on various sizes with different degrees of bend, have shown the area of the bend to be definitely stronger, failure always occurring away from the bend. Cold-bending has a greater strengthening effect than the weakening resulting from wall reduction.

Numerous other advantages of the tubular section, other than those previously mentioned, may assume importance in special applications. In outdoor structures, the effect of wind resistance is reduced to a minimum. Where space limitations are imposed, or portability is a necessity, tubing can be readily adapted to telescoping assemblies. The aircraft industry has utilized tubing in an infinite variety of capacities. By applying the principle of unit design, or section-stressed construction, and taking advantage of the strength properties of tubular elements, considerable savings in weight have been effected.

To withstand stresses in any specific direction, or provide necessary mounting facilities, plates or trusses may be used to reinforce a tubular section. The main bed of the weaving machine, *Fig.* 7, is made up from round tubing with longitudinal angles and flat end plates welded-on. Other applications of tubing in this machine include the six different rolls shown. Lightness, straightness, and dimensional accuracy requirements make tubing a necessity. The bottom roll, or cloth roll, is sometimes made of wood to meet purchase requirements but the roll made of tubing has approximately the same weight and entails about the same cost.

Materials: Versatility of the tubular section is further enhanced when advantage is taken of the spe-

cial properties of the steel analyses, anneals, and heat treatments available. Alloy tubing properties have found wide utility in bearings, axles for automobiles, trucks, and railroad cars, and many other applications in which their high-strength and workability are especially useful. The corrosion and heat-resistant characteristics of stainless steel also extend tubing applicability. For Lehr rolls used in glass manufacture, type 405 stainless tubes provide freedom from scaling at 1200 F. Stainless tubes have also been applied advantageously to large induced draft cooling tower fans for corrosive atmospheres. The blade design of *Fig.* 8 is essentially a single tubular-spar wing structure. The welded tubular frame forms a lightweight, high-strength unit for large 212 and 240-inch diameter fans.

Design and Tolerances: Considerations involved in producibility will vary somewhat according to the production facilities at hand. Tubing, because of its range of availability—analyses, shapes, sizes, etc.—provides a wide latitude of selection to meet varying requirements. It may be used as a raw material upon which subsequent fabrication operations are to be performed or it may be used as a finished product where possibly the only operation will be cutting to length.

In general, advantageous use of tubing for hollow parts falls into several categories. These classifications are:

1. Long parts requiring deep drilling and parts having large outside diameters and relatively thin wall sections.

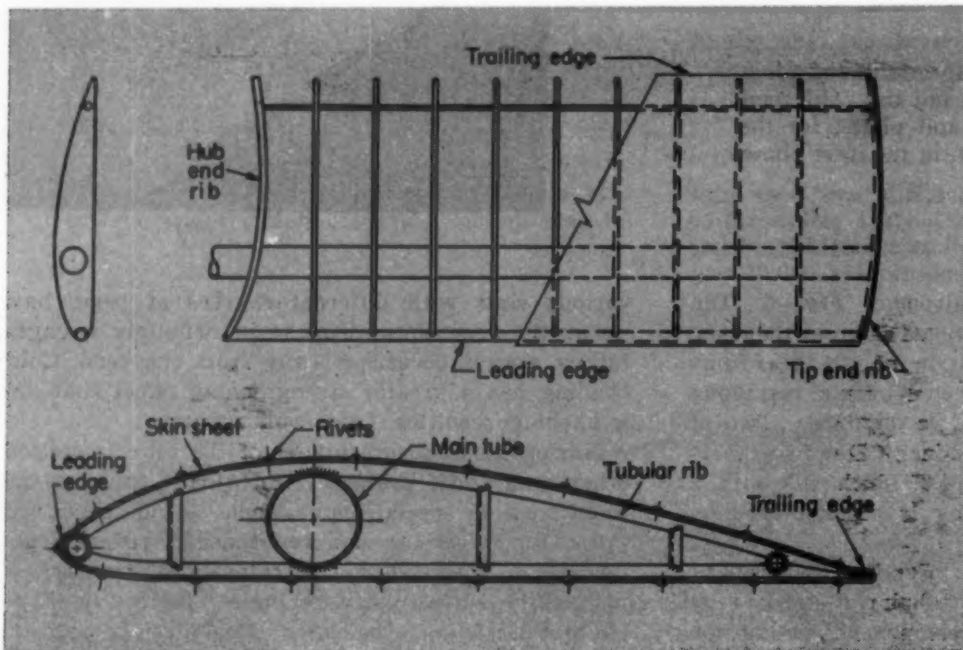
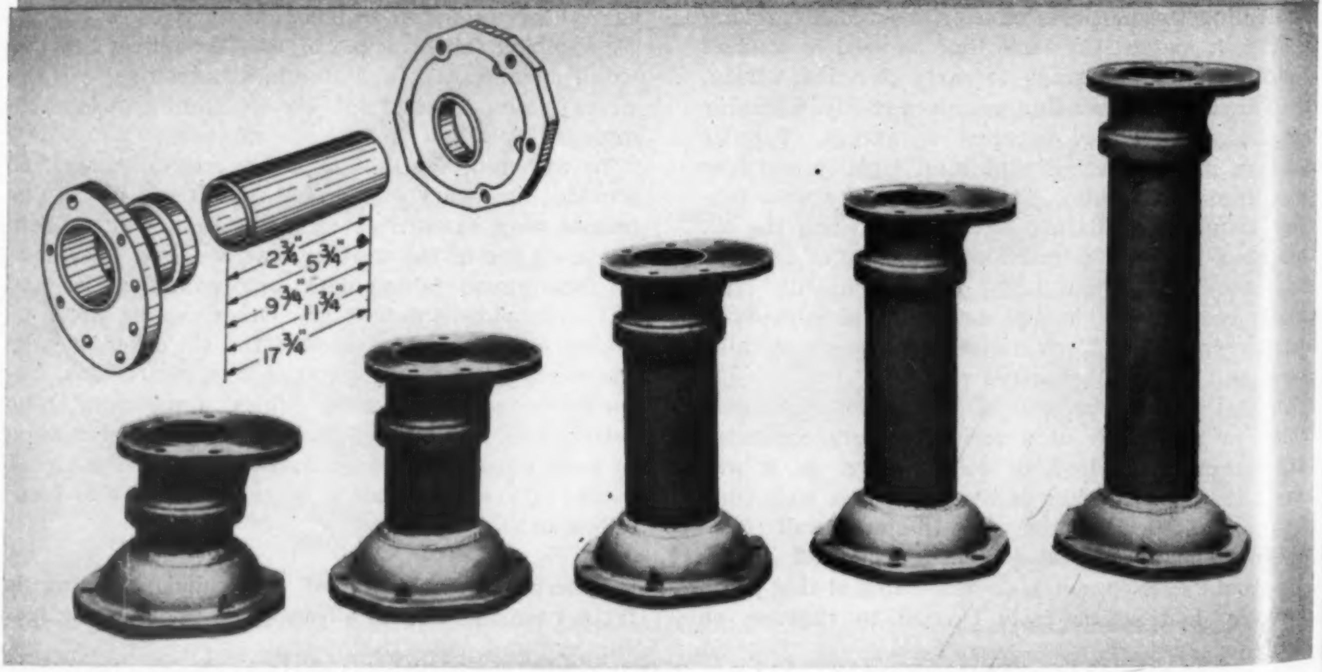


Fig. 8—Left—Large cooling tower fan blade is built up like a wing structure with tubular spars for strength and rigidity

Fig. 9 — Below — Design of tractor spacer units by Superior Steel & Malleable Castings Co., using cut to length tubular center sections to meet five different size requirements as shown, substantially reduced weight and cost of original one-piece casting



2. Parts for which tubing costs are equal to or more than other forms of material but savings in machining and other processing offset the additional material cost
3. Unusual designs where limitations imposed by available equipment make the use of tubing advantageous. Here, comparative cost of raw material would not be a major factor.

As originally designed to meet five different size requirements, the drive spacer units shown in *Fig.* 9 were expensive to produce as a one-piece steel casting. Separate core boxes and pattern equipment, expensive cleaning and handling procedures and extensive jig and fixture setups for each size left much to be desired. In addition, an excessive amount of critical steel was being used, the weight for the five

Fig. 10 — Above — Tubular cross-sections may be modified to meet a variety of design requirements. These parts of Rockrite tubing were shaped on a Federal-Westin hot forming machine

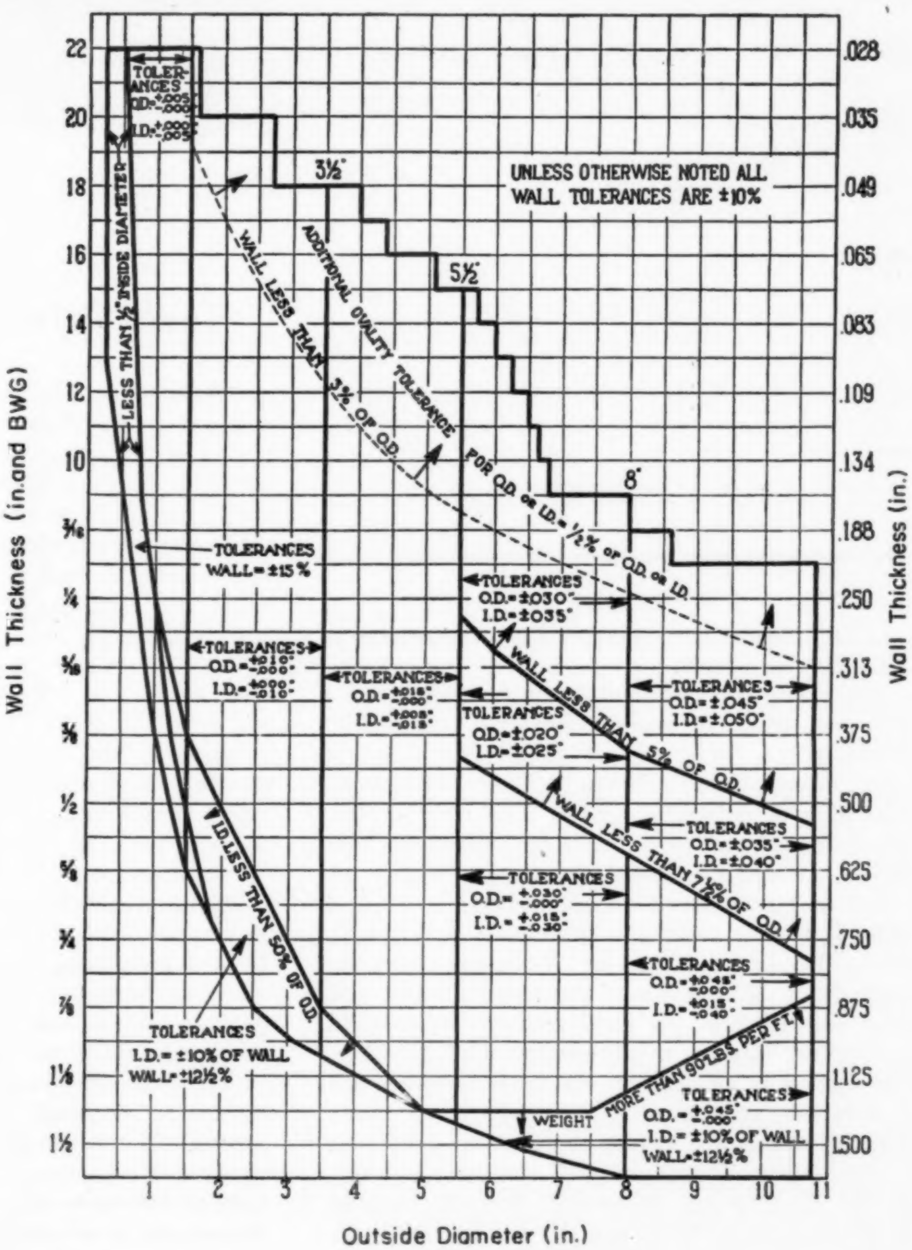


Fig. 11—Right—Chart showing typical tolerances for cold-drawn round seamless tubing

sizes averaging 52.5 pounds. The redesigned spacer employs two steel castings welded to a tubular center member cut to the length required. The tubular members not only provide the necessary strength, shock and fatigue resistance, but also resulted in substantial weight and cost reductions. Total cost of the part was reduced 38.4 per cent and weight 30 per cent. Only a single set of inexpensive patterns is required and cleaning and handling operations have been greatly simplified.

A shift sleeve for automotive use made from a forged ring required several separate machining operations with a resultant cost of $1.71 per finished part. By using a tubular section, 13 cents was saved on each sleeve in material costs and, since long lengths of tubing—unlike individual forgings—could be machined on a 4-spindle automatic screw machine, 11 cents per unit was saved on production.

Tubing is readily formed by a number of different methods. Beading, expanding, swaging, bending, spinning, flanging, upsetting, grooving, fluting, and tapering can be used to modify cross sections to meet different design requirements, *Fig.* 10. The tubular section is also easily adaptable to many different methods of joining.

Tube tolerances are applicable to two cross sectional dimensions. If OD and wall thickness are specified, the ID may not necessarily conform to published tolerances and if ID and OD are specified, the wall thickness may not necessarily conform. Typical tolerances for standard production-run cold-drawn round seamless tubing are shown in the chart, *Fig.* 11. Welded tube tolerances are shown in TABLE 1. These are only general size accuracies used with standard tubes. Wall thickness, diameter, etc., will alter these figures for specific sizes.

Table 1—Size Tolerances for Round Welded Tubing*

(cold-rolled carbon steel)

Tube Size (in.)	Wall Thickness (BWG)	Variations OD (in., plus or minus)	ID
¼ to ⅜	16 to 22	.003	.008
¼ to ⅜	14	.003	.010
½ to ⅝	20 to 22	.004	.005
½ to ⅝	16 to 18	.004	.005
½ to ⅝	12 to 14	.004	.009
¾ to 1⅛	18 to 22	.004	.005
¾ to 1⅛	14 to 16	.004	.005
¾ to 1⅛	11 to 13	.004	.008
1¼ to 2	18 to 22	.005	.006
1¼ to 2	14 to 16	.005	.006
1¼ to 2	9 to 13	.005	.008
2⅛ to 2½	18 to 20	.006	.007
2⅛ to 2½	14 to 16	.006	.007
2⅛ to 2½	9 to 13	.006	.009
2⅝ to 3	18 to 20	.010	.012
2⅝ to 3	14 to 16	.008	.010
2⅝ to 3	9. to 13	.008	.010
3⅛ to 3½	16 to 18	.010	.012
3⅛ to 3½	9 to 14	.008	.012
3⅝ to 4	14 to 16	.010	.014
3⅝ to 4	8 to 13	.010	.016
4⅛ to 5	14 to 16	.020	.020
4⅛ to 5	8 to 13	.015	.018

* Accuracies shown are for standard tube—Diameters and wall thicknesses for specific sizes may alter these figures.

ACKNOWLEDGEMENT

MACHINE DESIGN acknowledges with appreciation the co-operation of the following organizations in the preparation of this article:

Agaloy Tubing Co. .... Springfield, O.
Allegheny Ludlum Steel Corp. .... Pittsburgh, Pa.
American Iron and Steel Institute .... New York, N. Y.
Armco Steel Corp. .... Middletown, O.
Babcock and Wilcox Co., Tubular Products Div., ... Beaver Falls, Pa.
Carpenter Steel Co., Alloy Tube Div. .... Union, N. J.
Caterpillar Tractor Co. .... Peoria, Ill.
Formed Steel Tube Institute .... Cleveland, O.
Formed Tubes Inc. .... Sturgis, Mich.
Globe Steel Tubes Co. .... Milwaukee, Wis.
Jones and Laughlin Steel Corp. .... Pittsburgh, Pa.
Lincoln Electric Co. (*Fig.* 1) .... Cleveland, O.
Marley Co. (*Figs.* 2, 3 and 8) .... Kansas City, Mo.
National Tube Div., U. S. Steel Co. (*Fig.* 11) .... Pittsburgh, Pa.
Ohio Seamless Tube Co. .... Shelby, O.
Pittsburgh Steel Co. .... Pittsburgh, Pa.
Steel Founders' Society (*Fig.* 9) .... Cleveland, O.
Superior Tube Co. .... Norristown, Pa.
Timken Roller Bearing Co., Steel and Tube Div., ... Canton, O.
Tube Reducing Corp. (*Figs.* 5 and 10) .... Wallington, N. J.
Wallace Supplies Mfg. Co. (*Fig.* 6) .... Chicago, Ill.
Warner & Swasey Co. (*Fig.* 7) .... Cleveland, O.

Chain Drive Center Distance

CORRECTION: In the article "Polygonal Action in Chain Drives" (MACHINE DESIGN, September, 1952) the equation given for determining the optimum center distance would yield a short or less-than-optimum center distance. Correctly expressed, the distance between sprocket centers for minimum polygonal action is

$$C = \sqrt{L^2 + \frac{P^2}{4}\left(\cot\frac{180}{N} - \cot\frac{180}{n}\right)^2}$$

where L = length of tangent span, inches; P = pitch of chain, inches; N and n are the numbers of teeth in the large and small sprockets, respectively; and the angles $180/N$ and $180/n$ are in degrees.

"Human incentives are simple, and relatively few in number. Most of us derive personal satisfaction from a knowledge that we have done our best, and some would consider this inner satisfaction as adequate reward. Some strive for the prestige success will bring, for the admiration and respect of their fellowmen. Some work for power and the influence so obtained over the lives and activities of others. But for most of us, I think we will agree that the strongest and most desirable incentive of all is financial gain—not, of course, in money itself, but because of what one can do with it."—CRAWFORD H. GREENEWALT, *president, E. I. duPont de Nemours & Co.*

By W. A. Tuplin
Professor of Applied Mechanics
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SIMPLIFIED Gear Design

... deals with the tangible properties of generated gear teeth and provides flexibility with standard pitches

USUAL textbook presentations of gear tooth geometry for generated involute gears working at nonstandard center distances have deterred many designers because of the complications which seemed to be involved. However, rational examination of the essentials of the problem shows that these complications are superfluous. The object of the design procedure presented in this article is to achieve the utmost simplicity while still offering such flexibility that almost every problem in gear design can be solved without resorting to hobs or cutters of nonstandard pitch. There need be no exception at all for helical gears, and only in rare cases for spur gears—for example, where the number of teeth in the pinion must be very small.

Although there is no single procedure which is widely accepted for the design of gears to work at what may be called nonstandard center distances, the usual course is to fix the diameter of the pinion according to the number of teeth, the pitch of the cutter and some correction system, considering the pinion as if it were meshing at a standard center distance. The outside diameter of the gear is made greater or less than the standard amount by twice the difference between the working and standard center distances. Instructions are given for calculating the pitch diameters of engagement and for determining the pressure angle of engagement, which is used in conjunction with involute functions to find out how much the cutting depth of the gear must differ from standard to produce no backlash. It is usually recommended that the tooth form of the gear should be set out on the drawing board to help in deciding whether its tip width or its approach to undercutting are acceptable.

Distinction between standard and nonstandard center distances for involute spur or helical gears is purely artificial, and in the system of design described here is not recognized. Although it is conventional to say that the designed center distance of a pair of gears is half the sum of the pitch diameters, it is well known that an involute gear has no actual pitch diameter. This inconsistency is avoided by specifying that the designed center distance is half the sum of

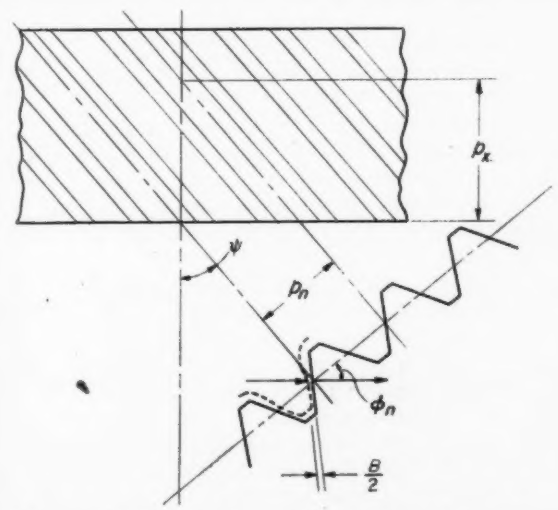


Fig. 1—Basic helical gear dimensions shown include helix angle ψ, axial pitch p_x, normal pitch p_n, normal pressure angle ϕ_n, and normal backlash B

the tip diameters less the standard working depth of the teeth, which gives

$$C = \frac{d_o + D_o}{2} - \frac{2}{P_n} \qquad (1)$$

Factors are defined in the Nomenclature and *Figs.* 1 and 2.

Equation 1 refers not to an imaginary diameter, but rather to a directly measurable quantity of each gear, the diameter of its tip-circle. The essential dimensions of the teeth at this stage are the working depth, $2/P_n$, and the full depth, taken here to be at least $2.25/P_n$, *Fig.* 2. Limitations discussed here and underlying *Fig.* 3 refer to a normal pressure angle of 20 degrees; however, the same general methods are applicable to teeth of any nominal pressure angle.

For helical gears, the approximate value of the helix angle, ψ, is selected to suit operating conditions, while the exact value is selected to make p_x, which is equal to p_n cosec ψ, expressible as a ratio of small whole numbers, TABLE 1*. Pinion and gear outside diameters may be obtained from

$$d_o = \frac{N_P \sec\psi + 3}{P_n} \qquad (2)$$

$$D_o = \frac{N_G \sec\psi + S}{P_n} \qquad (3)$$

In these equations, the outside diameter is equal to the pitch diameter of generation, $(N \sec\psi)/P_n$, plus twice an "addendum" which is $3/P_n$ in Equation 2 and S/P_n in Equation 3—a contrast to the more conventional form in which the addendum is taken as $2/P_n$ for both gears or, alternatively, pinion and gear are given different addendums, totalling $4/P_n$.

In the design stages, S may be given any value in the range specified in *Fig.* 3 for value of N_G and ψ. If the designed center distance is fixed, the value of S is determined by Equation 13. If the designed center distance is not closely restricted, there is a slight preference to give S the nearest permitted value to 1.

For the special case of equal diameter gears, $d_o = D_o$, Equation 3 should be used for calculations. Again, S may be given any value permitted in *Fig.* 3. If the designed center distance is not closely restricted, there is a slight preference for giving S the nearest permitted value to 2.

Lead distances may be calculated from

$$l_P = N_P p_x \qquad (4)$$

$$l_G = N_G p_x \qquad (5)$$

Dimensions of tooth thickness and height, shown in *Fig.* 2, may be determined from

$$g = \frac{1.708}{P_n} \qquad (6)$$

$$h = \frac{1.189}{P_n} \qquad (7)$$

$$G = \frac{0.321S + 0.745}{P_n} \qquad (8)$$

$$H = \frac{0.442S - 0.137}{P_n} \qquad (9)$$

These tooth thicknesses correspond to cutting full standard depth and provide a normal backlash of

$$B = \frac{0.65\,(S-1)^2\,\psi}{CP_n^2} \qquad (10)$$

For additional backlash, dB, the thickness of each gear should be decreased the amount $(dB \sec\phi_n)/2$ which equals $0.53dB$ when ϕ_n is 20 degrees. The corresponding excess over standard depth is $(dB \operatorname{cosec}\phi_n)/4$ or $0.73dB$ if $\phi_n = 20$ degrees.

To secure the condition of no backlash at the designed center distance, the cutting depth must, in general, be less than standard. In such cases, the bottom clearance is also less than standard but is still adequate for all practical purposes. There is no need for special reduction in tip diameter (or topping) that is often specified in such circumstances.

Slide-rule calculation is sufficiently accurate for this method of design provided that the final value of D_o is determined from

$$D_o = 2C - d_o + \frac{4}{P_n} \qquad (11)$$

Slide-rule calculation of D_o by Equation 3 may not exactly agree with result obtained with Equation 11, in which case the value given by Equation 3 should be disregarded. Any considerable discrepancy between values of D_o indicates some arithmetical error.

If center distance, numbers of teeth and helix angle have been tentatively fixed, the approximate value of P_n can be determined from

$$P_n \approx \frac{1}{2C}\left[(N_P + N_G)\sec\psi + 1\right] \qquad (12)$$

By appropriate selection of the helix angle, ψ, values

* W. A. Tuplin—"What Helix Angle?," MACHINE DESIGN, April 1951, Page 173.

Nomenclature

B = Normal backlash, inches
C = Center distance of gears, inches
dB = Incremental change in backlash, inches
d_b = Base diameter, pinion or gear, inches
dh = Incremental change in tooth depth, inches
d_o = Outside diameter of pinion, inches
D_o = Outside diameter of gear, inches
g = Thickness of pinion tooth, inches
G = Thickness of gear tooth, inches
h = Height of pinion tooth, inches
H = Height of gear tooth, inches
l_P = Lead of pinion, inches
l_G = Lead of gear, inches
N_P, N_G = Number of teeth in pinion or gear
p_x = Axial pitch of pinion and gear, inches
p_n = Nominal normal circular pitch, inches
P_n = Nominal normal diametral pitch
S = Quantity defined by *Fig.* 1.
ϕ_n = Normal pressure angle, degrees
ψ = Helix angle, degrees
z = Constant

of P_n may be obtained which closely approximate standard values. The same flexibility is not available in spur gears since $\psi = 0$. However, to best present the method of design described here, no use will be made of this possibility in angle selection.

When a tentative value of P_n has been found, S may be calculated from

$$S = 2CP_n - (N_P + N_G) \sec \psi + 1 \quad \ldots\ldots\ldots\ldots (13)$$

If the value obtained from this equation falls within the range permitted by *Fig.* 1 for the particular values of N_G and ψ, the proposed value of P_n and the calculated value of S may be adopted. Dimensions of the gears may then be calculated from the various equations previously enumerated.

Application of Equations: As an example, determine the dimensions of a pair of helical gears to operate at a center distance of 15 inches and a ratio of about 23/63. The teeth are to have a helix angle of 31 deg 34 min and are to be cut by a hob of standard normal diametral pitch. Axial pitch for this helix angle is $6/P_n$, TABLE 1.

From Equation 12, $P_n \approx [(23 + 63)\ 1.172 + 1]/2\ (15) = 3.39$.

The nearest standard diametral pitch is $P_n = 3.5$. Using this value in Equation 13, $S = 2\ (15)\ (3.5) - 86\ (1.172) + 1 = 5.2$.

Referring to *Fig.* 3, this value of S is greater than is permissible for $N = 63$ and $\psi = 31$ deg 34 min. It is therefore necessary to adopt either a coarser pitch than $P_n = 3.5$ or to increase the numbers of teeth. The former is impractical because of the considerable step from a pitch of 3.5 to the next standard pitch of 3. The alternative is to increase the numbers of teeth approximately in the ratio of 3.5/3.39, which means $N_P = 24$ and $N_G = 66$. Tentatively adopting this basis, and using Equation 13, $S = 2\ (15)\ (3.5) - 90\ (1.172) + 1 = 0.5$.

This also is inadmissible. Next, trying $N_P = 24$ and $N_G = 65$ instead of 66, $S = 2\ (15)\ (3.5) - 89\ (1.172) + 1.67 = 1.67$.

This value of S is within the permitted range and may be used.

From Equation 2, $d_o = [24\ (1.172) + 3]/3.5 = 8.90$ inches.

From Equation 3, $D_o = [65\ (1.172) + 1.67]/3.5 = 22.25$ inches.

From Equation 1, $C = (8.90 + 22.25)/2 - 2/3.5 = 15.004$ inches. Since this does not give an exact result of $C = 15.000$, the value of D_o is derived from Equation 11 to give $D_o = 2\ (15) - 8.90 + 4/3.5 = 22.243$ inches as the exact solution.

Using these accepted values, the other quantities may now be evaluated.

By definition, $p_n = \pi/P_n$ and $p_x = (\pi \text{ cosec } 31 \text{ deg } 34 \text{ min})/P_n = 6\pi/3.5\pi = 12/7$ inches.

From Equation 4, $l_P = 24\ (12)/7 = 288/7$ inches.

From Equation 5, $l_G = 65\ (12)/7 = 780/7$ inches.

From Equation 10, $B = [0.65\ (0.67)^2\ (\cos^2 31 \text{ deg } 34 \text{ min})]/15\ (3.5)^2 = 0.0012$ inches, which is normal backlash with full depth teeth.

From Equation 6, $g = 1.708/3.5 = 0.488$ inches.

From Equation 7, $h = 1.189/3.5 = 0.339$ inches.

From Equation 8, $G = [0.321\ (1.67) + 0.745]/3.5 = 0.366$ inches.

From Equation 9, $H = [0.442\ (1.67) - 0.137]/3.5 = 0.172$ inches.

If zero backlash is desired, each tooth thickness should be increased by $0.0012\ (0.53) = 0.0006$ inches. This makes each tooth depth less than the standard amount by $0.0012\ (0.73) = 0.0009$ inch. These computations for eliminating backlash are given merely to illustrate the procedure and are not meant to suggest that such fine control of tooth thickness is easily attainable by the ordinary gear cutting processes.

Analysis of Equations: In the method of design described, a so called correction system has been used and gears have been proportioned to work at nonstandard center distances. Neither of these terms is necessary, and as no pitch circle is mentioned, it is impossible to specify an addendum. All of these designations date back to the cycloidal system of toothed gearing and it has taken a long time to realize how useless these encumbrances are in a method of design based on tooth generation by standard cutters.

This method is much simpler than the normally recommended design procedure for nonstandard gears; in particular, the involute function is not even mentioned. Also, slide-rule calculation is sufficiently

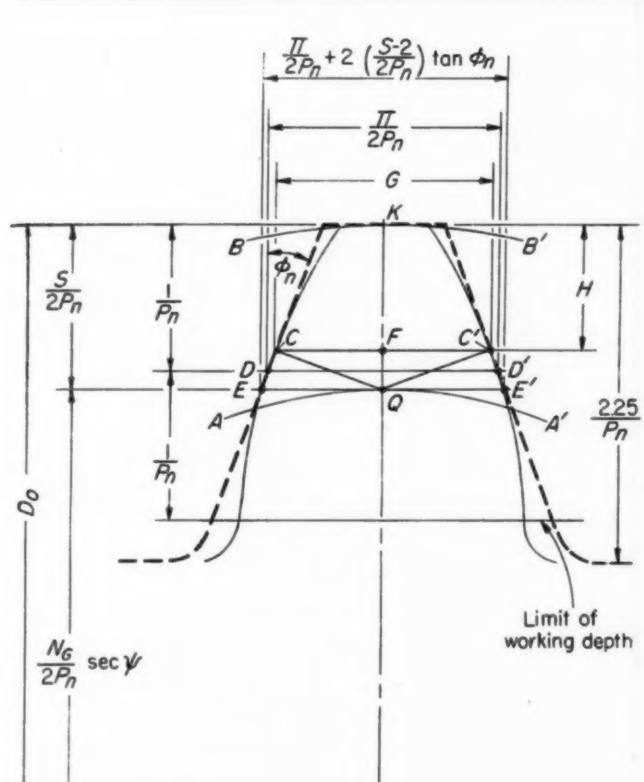


Fig. 2—Basic dimensions of simplified system of design include thickness tooth *G*, and height *H*. Conventional addendum and pitch diameter are disregarded

accurate for determining essential dimensions.

The simplicity of this method has been found quite disconcerting by some gear technicians who have been accustomed to enduring the conventional complications. However, the important and extremely useful fact that seems to have been forgotten is that for a given number of teeth, normal pitch and helix angle a gear may have any outside diameter within a certain range. With outside diameters not in that range, the tooth form may be unsatisfactory either because it is weakened by undercutting or has an undesirably narrow crest. Limits within which satisfactory tooth shapes are produced are defined in *Fig.* 3, and, although these ranges are not everywhere as wide as they might be, they avoid any need for a nonstandard cutter for any application. Limiting values of *S* have been determined empirically to avoid sharp cresting of the teeth and undercutting.

Although the normal pressure angle of the generating cutter is 20 degrees, the working pressure angle, even of a spur gear, is not necessarily the same, but departure from that figure never causes bearing loads to be appreciably greater than those corresponding to a 20 degree normal pressure angle. Anyone who feels the need of reassurance on this point may, of course, always calculate the working pressure angle by well known textbook procedures, and may determine the tooth form by calculation or otherwise. There is, however, no practical need for this determination, as it has all been done many times before.

With straight spur gears, the load capacity for given materials, gear diameter and tooth face width tends to rise with increase in working pressure angle. If it is desired to take advantage of this extra capacity, and if the center distance is not closely restricted, the value of *S* should be the largest permitted by *Fig.* 3. The "slight preference" mentioned after Equation 3 is overridden in this case, but it still applies to helical gears because their load capacity is independent of pressure angle. The working pressure angle,

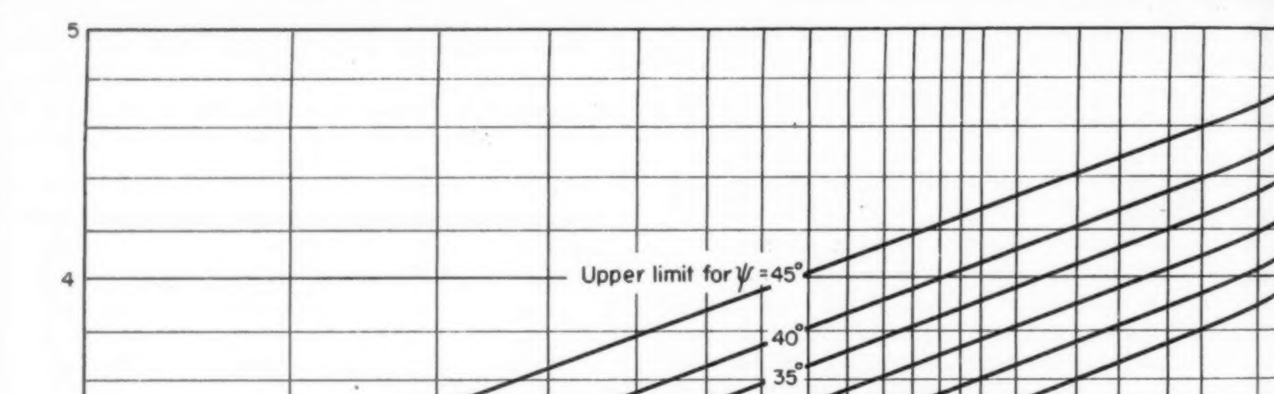


Fig. 3—Limiting values of S, determined empirically to avoid sharp cresting and undercutting, provide adequate range in design to eliminate nonstandard pitches

ϕ_w, of spur gears for this method of design is

$$\phi_w = \cos^{-1}\left[0.940\left(\frac{N_P + N_G}{2P_n C}\right)\right]$$

where $0.940 = \cos 20$ deg.

In the method of measuring tooth thickness given here, the formulas refer to measurement between points at which the tangent planes are inclined to each other at 40 degrees. For a given normal pitch and helix angle, this dimension depends only on its distance from the plane at which the rack tooth thickness is half the pitch, and not on the number of teeth in the gear. In *Fig.* 2, O represents the axis of the gear, the arc AA' part of the pitch circle of generation and arc BB' part of the tip circle. A tooth is placed symmetrically in relation to the point (Q) of contact of the pitch circle of engagement of the gear and the pitch line of engagement of the imaginary rack whose points of contact with the gear tooth are determined by drawing perpendiculars QC and QC' from Q to the flanks of the rack teeth. The space width of the rack is half the pitch at a distance $1/P_n$ from the inner and outer limits of the working depth. Consequently, at a radial distance $1/P_n$ from the tip circle of the gear, the space width DD' of the rack is $\pi/2P_n$. At the distance, $S/2P_n$, of the pitch circle of generation from the tip, the space width differs from DD' by the product of 2 tan ϕ_n and the difference between $1/P_n$ and $S/2P_n$.

From the geometry of *Fig.* 2:

$$EE' = DD' + 2\tan\phi_n\left(\frac{S-2}{2P_n}\right)$$

$$= \frac{1}{P_n}\left[\frac{\pi}{2} + (S-2)\tan\phi_n\right]$$

$$G = CC' = 2CQ\cos\phi_n = 2EQ\cos^2\phi_n$$

$$= \frac{\cos^2\phi_n}{P_n}\left[\frac{\pi}{2} + (S-2)\tan\phi_n\right]$$

If $\phi_n = 20$ degrees, Equation 8 is obtained. Also,

$$FQ = CQ\sin\phi_n = EQ\sin\phi_n\cos\phi_n$$

$$H = KF = KQ - FQ$$

$$= \frac{S}{2P_n} - \frac{\sin\phi_n\cos\phi_n}{2P_n}\left[\frac{\pi}{2} + (S-2)\tan\phi_n\right]$$

If $\phi_n = 20$ degrees, Equation 9 is obtained.

For the pinion, S has the value 3. Inserting this value in the expressions for tooth thickness and height of tip above the plane of measurement gives Equations 6 and 7.

If the depth of cutting is changed by an incremental amount dh, the distance between the two positions of the basic rack, measured perpendicular to one flank, is $dh \sin\phi_n$, and if the same change is made in the mating gear the increase in normal backlash is $4\,dh\sin\phi_n$. Therefore to change the backlash by an amount dB the cutting depth of each gear must be changed by $dh = dB/(4\sin\phi_n) = 0.73dB$ if $\phi_n = 20$ degrees. The corresponding change in tooth-thickness measured from C to C' is $2\,dh\tan\phi_n = (dB\sec\phi_n)/2 = 0.53dB$ if $\phi_n = 20$ degrees.

The scheme described in this article applies to involute gears with 20 degree normal pressure angle and working depth equal to $2/P_n$. The same principles can be applied to involute gears with different forms of basic rack; the only essential differences in design procedure are in the permitted range of S and (where the pressure angle is varied) in the tooth thickness and height. These quantities, as defined for two alternative rack forms, are:

1. For a normal pressure angle of 20 degrees and a working depth of z/P_n:

$$S_{max} = 5.25 - 0.625\,z - \frac{10\cos^2\psi}{N_G}$$

$$S_{min} = z - 1 + \frac{20\cos^2\psi}{N_G}$$

$$g = \frac{2.350 - 0.321\,z}{P_n}$$

$$h = \frac{1.072 + 0.0585\,z}{P_n}$$

$$G = \frac{1.387 + 0.321\,(S - z)}{P_n}$$

$$H = \frac{0.442\,S + 0.0585\,z - 0.254}{P_n}$$

2. For a normal pressure angle of 14½ degrees and a working depth of $2/P_n$:

$$S_{max} = 4 - \frac{16\cos^2\psi}{N_G}$$

$$S_{min} = 1 + \frac{32\cos^2\psi}{N_G}$$

$$g = \frac{1.714}{P_n}$$

$$h = \frac{1.280}{P_n}$$

$$G = \frac{0.242S + 0.988}{P_n}$$

$$H = \frac{0.469S - 0.127}{P_n}$$

$$B = \frac{0.94\,(S-1)^2\cos^2\psi}{P_n^2}$$

For extra backlash dB, the tooth thickness of each gear should be reduced by $0.52dB$. The corresponding excess over standard depth is dB.

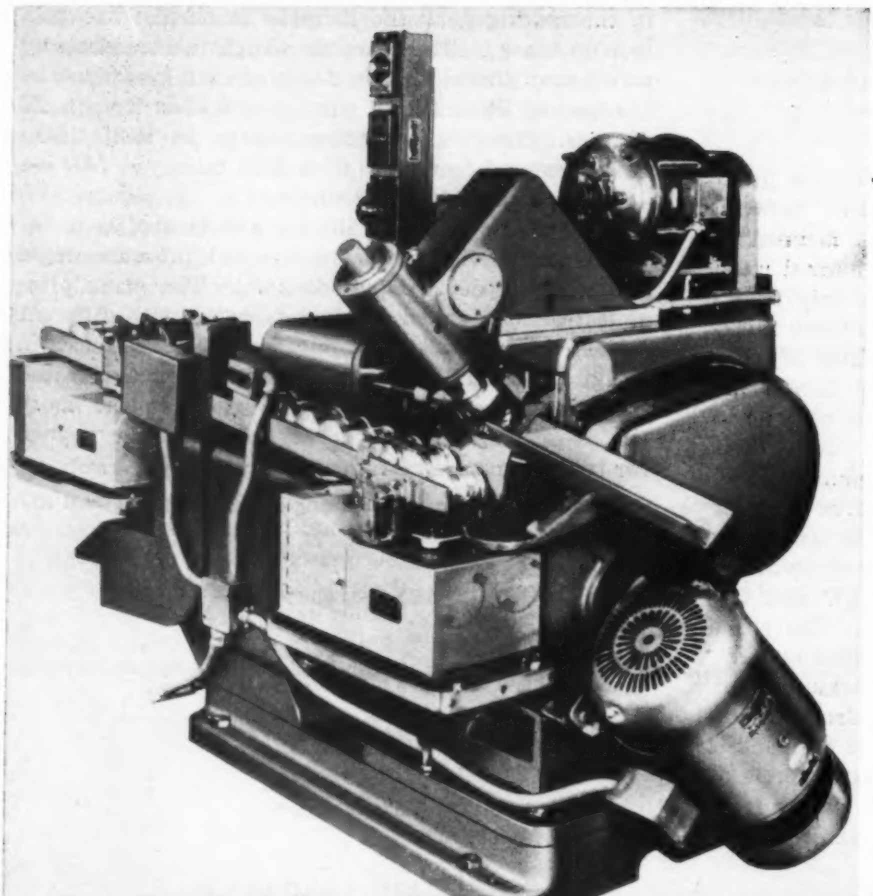
Base diameter may be calculated from

$$d_b = \frac{N}{\sqrt{\left(P_n\sec\phi_n - \dfrac{\pi}{p_x}\right)\left(P_n\sec\phi_n + \dfrac{\pi}{p_x}\right)}}$$

Although this dimension is not required for manufacturing purposes, it may be needed to set up certain types of tooth-profile testing machines.

Table 1—Preferred Helix Angles

Normal Pitch	Axial Pitch	Helix Angle
Standard Circular Pitch	4 p_n	14° 30′
	2.5 p_n	23° 30′
	2 p_n	30°
Standard Diametral Pitch	$12/P_n$	15° 12′
	$8/P_n$	23° 10′
	$6/P_n$	31° 34′

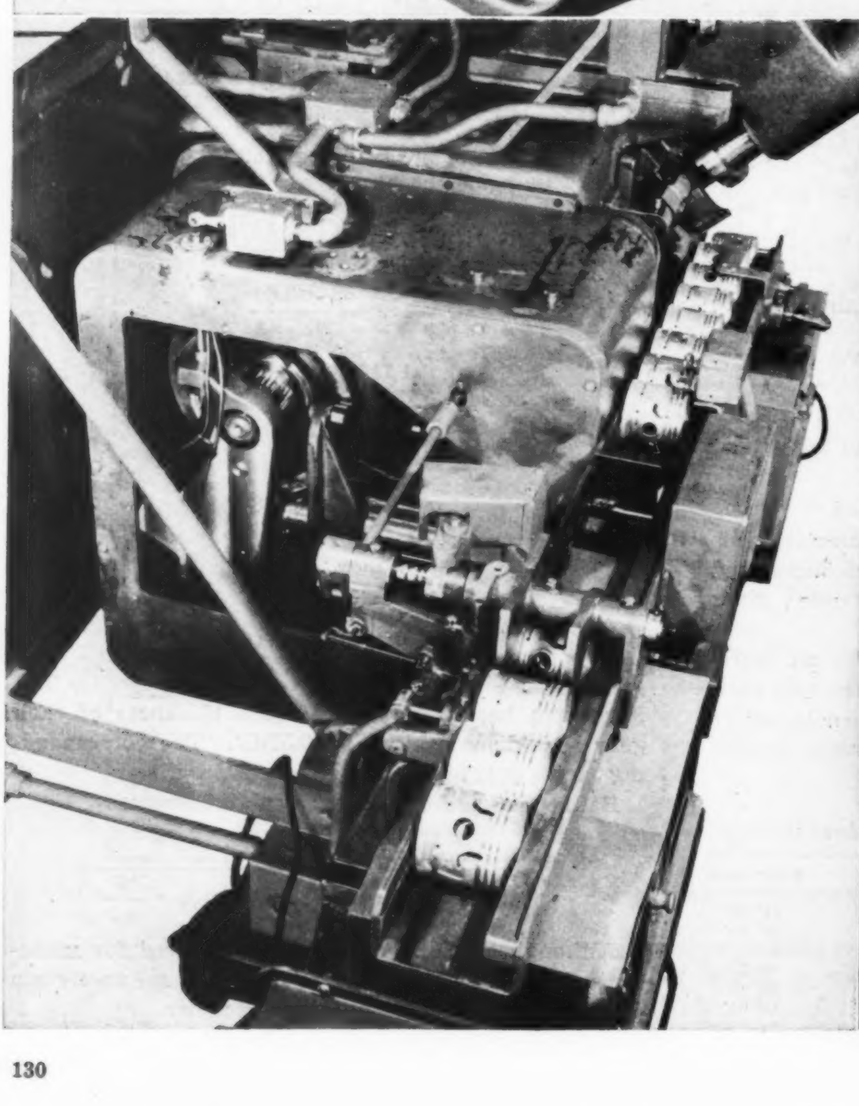


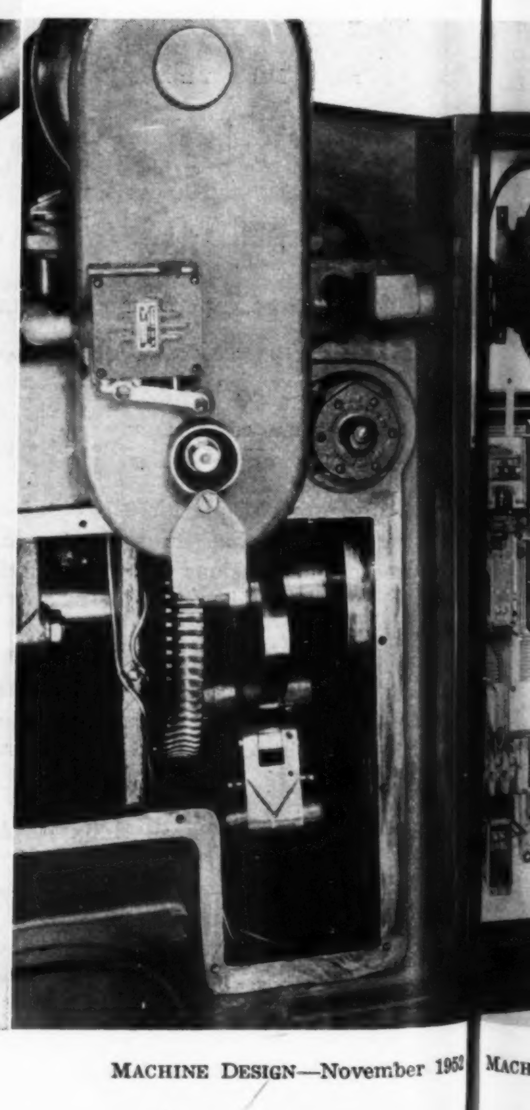
CONT

Piston Bala

FINISHED weight of automotive pistons is established within plus or minus 0.035-oz in fully automatic production by the new Morris piston balancer, left. Without operator handling, pistons enter the balancer from a conveyor line, are corrected for weight by end-milling metal from trim pads inside the skirt and then are discharged onto a distributing conveyor.

Operation of the balancer is electromechanical throughout, with cams and linkages co-ordinating the various motions during cycles





PORARY DESIGN

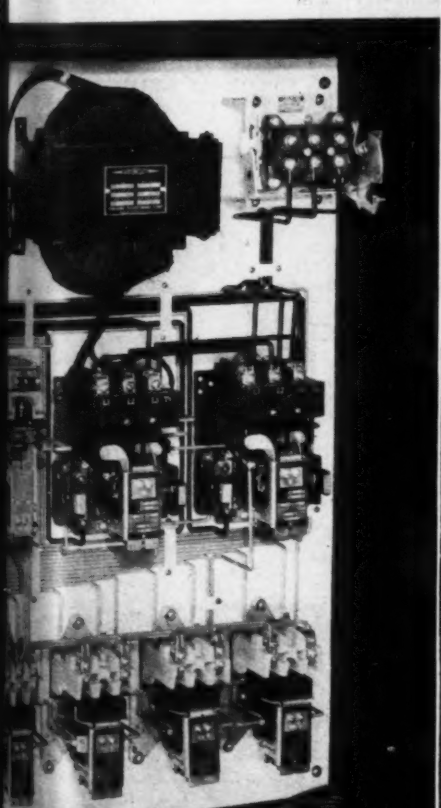
s Weight Pads

ranging between 5 and 8 seconds length. On entering the infeed chute, proper orientation of the pistons is determined by an interference finger which extends through and inside one sidewall of the chute. If in reverse position, the piston is delayed by the finger until manually removed from the chute and repositioned end-for-end. Properly oriented pistons are indexed for preliminary weighing, below left, onto the platform of a Shadograph beam scale.

In this weighing operation, which constitutes an automatic inspection for proper trim-pad machining allowance, pistons of abnormal and subnormal overweight are detected for subsequent discharge into either of two discharge chutes beyond the scale for further inspection. Trapdoors over the overweight and underweight chutes are solenoid-controlled from the weight grading scale.

If the weight of the piston is within prescribed limits, both trapdoors remain closed and the piston advances toward the balancing station.

An escapement mechanism, photo below, delivers the preweighed and properly oriented piston to a second Shadograph scale. Beam oscillations caused by the piston rolling into the scale V-blocks are damped by a built-in dashpot. When the beam comes to rest it is clamped in position to record the weight of the piston. By electrical control, recorded overweight is translated into terms of milling-cut depth required to remove the excess metal (weight) from the trim pads. Depth of cut is established with ref-



erence to the bottom of the piston skirt.

After "weighing-in" at the recording scale, the piston is lifted out of the scale V-blocks by a transfer arm and positioned on a locating ring against which it is held centrally by an overhead collet. An inserted-tooth cutter advances upward to the predetermined depth and trims off the excess metal. Chip removal from the trimming operation is by suction-aided gravity flow through the locating ring into a disposal duct. When the cutter head recedes, a mechanical hand advances from the side and transfers the balanced piston from the chuck to a discharge trough through which it passes to a conveyor.

The control mechanism of the piston balancer is powered through a combined V-belt and wormgear drive from a 1.5-hp motor, center, previous pages. The worm of this drive may be crank-operated with safety for setup purposes. Access to the worm is gained through a flap-covered hole in the belt guard. Swinging the flap aside to permit insertion of the crank opens a limit switch normally held closed in the motor circuit. This prevents accidental motor operation while the crank is in use.

Fishing Reel Funnels Line

QUITE different from ordinary fishing reels is the new Shakespeare No. 1850 spinning reel, below. Characteristic of this new design is the coaxial position of the reel spool with respect to the rod instead of the ordinary crossed-axes arrangement. Operation of this type reel is unique in that normally the line spool is stationary, whether casting or retrieving.

When casting, the line spirals freely from the stationary spool inside an outer cone-shaped housing. Through the nose of this housing the line "funnels" out in straight-line fashion without whip. Noteworthy in operation of the spinning reel is the absence of reel inertia which ordinarily requires sensitive control to prevent backlash and fouling of the line. Casting distance is determined simply by finger-snubbing the line itself during runout while the reel mechanism is locked in position to clear the line.

Driven by crank rotation through a 4 to 1 ratio bevel gear speed increaser, an inner cone assembly automatically picks up the line and, through simultaneous rotation and reciprocation, level-winds it onto the spool. Reciprocation is imparted to the inner cone by a scotch yoke arrangement designed into the crown of the bevel drive gear at the end of the crankshaft inside the main housing.

The line is picked up by a round Carboloy finger which normally is retracted inside the inner cone. With crank rotation the pickup finger swings outward through an aperture in the wall of the inner cone and sweeps around inside the outer housing. Two round wear strips, also of Carboloy grade 883 are brazed across the ends of the aperture to resist abrasion from the line as it is guided over the inner cone and onto the spool. Tension or tautness of the

line is controlled by axial friction braking against one flange of the spool. Friction material is a felt ring shouldered in a threaded and knurled adjusting ring which also supports the outer cone through a slip fit. Contacting end serrations on this ring, a spring finger maintains adjustment of the brake.

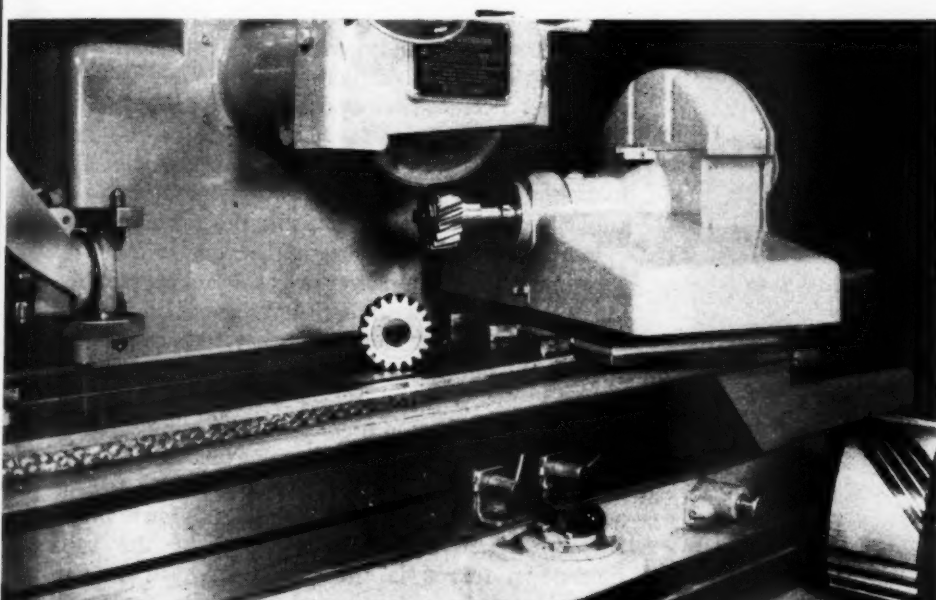
Made principally of aluminum, the new model casting reel weighs but eleven ounces. The aluminum parts are anodized prior to final finishing in forest green enamel. Line spools are interchangeable and require no special tools for removal. These spools range in capacity from 70 yards of 0.014-inch diameter to 300 yards of 0.008-inch diameter Nylon line.

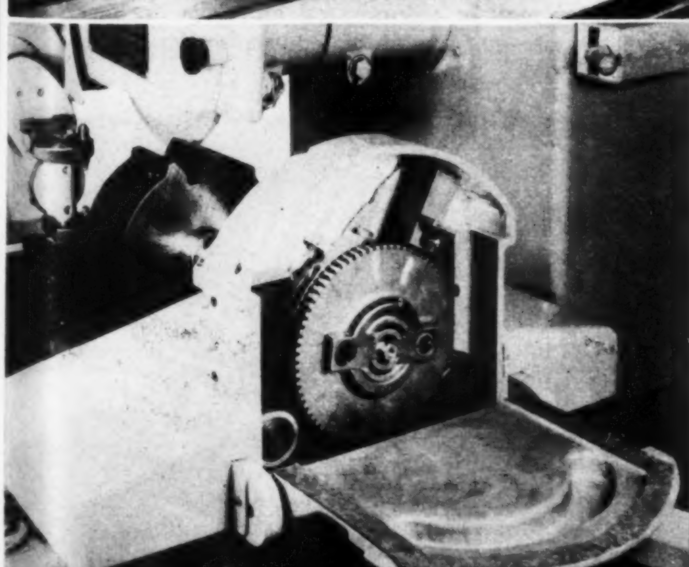
Steel Tape Drives Gear Grinder

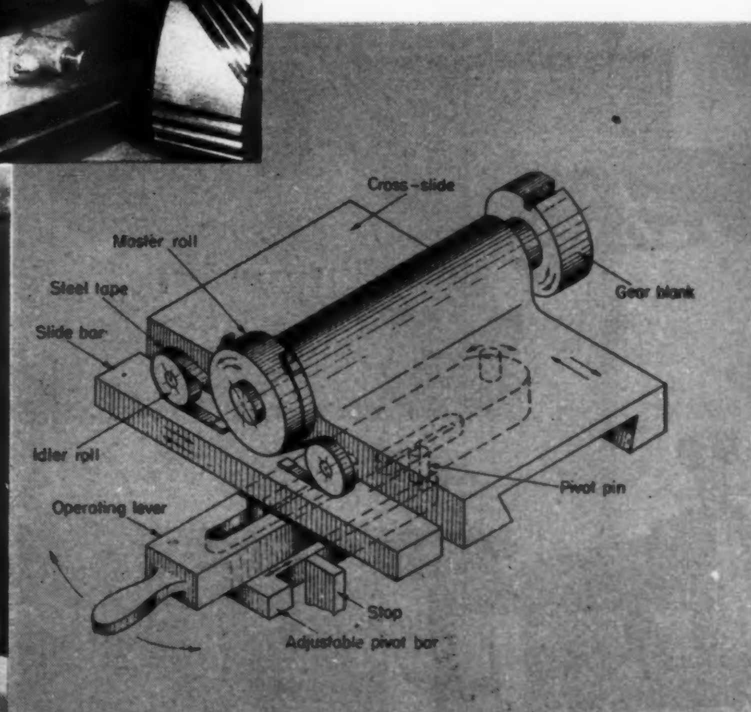
A NEW and unique gear grinding attachment which enables generating a variety of involute and other forms on a surface grinder, below, utilizes a steel-tape drive in conjunction with a cross-slide linkage to generate the forms. At one end, the two tapes employed are anchored to a master cylinder or roll, bottom photo. Carried on top of this roll is a spring-loaded finger which, in conjuction with an index plate on the fixture spindle, establishes circular spacing of the gear teeth. Index plates are interchangeable.

From the master roll, the tapes loop around idler rolls and are fastened at the other ends to a manually reciprocated slide bar; see sketch below. A stud on the underside of this bar engages a slotted operating lever which is fulcrumed on an adjustable pivot. The operating lever is also pivoted to the underside of the cross-slide so that when it is oscillated between fixed stops, co-ordinate oscillation of the spindle and transverse reciprocation of the cross-slide result. These motions, in conjunction with a grinding wheel formed to basic rack-tooth proportions, generate involute teeth in solid or preformed blanks as the wheel is fed to depth during table traverse.

For various diameter gears, the adjustable pivot is located simply by a gage block combination equal



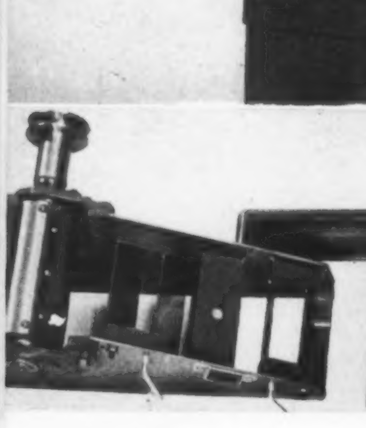


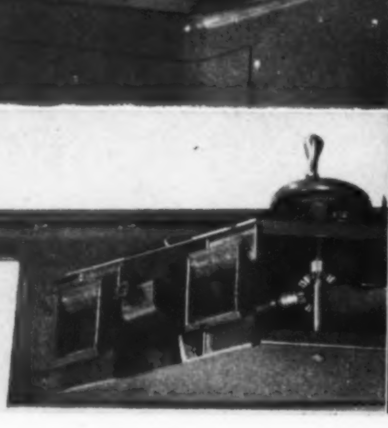


in length to one-half the gear pitch diameter. A ground strip along the base of the cross-slide facilitates positioning the generator angularly for producing helical gears.

This new DoALL development enables generating gears of any pressure angle, of 2 to 100 diametral pitch, and from one-half to six inches diameter on a low quantity basis at minimum tool investment. Involute spline members and chain sprockets also may be produced with the universal gear generator.

Welded Design Improves Jointer





LIGHTER weight, greater rigidity, and more permanent accuracy are achieved through design for welded-steel construction in the new Boice Crane jointer, left. Material and labor savings also are realized out of the development since the weldments are designed for production welding in jigs to minimize stock allowances for finish machining. Several small steel castings are employed in conjunction with the formed members to avoid complex machining.

Of particular importance is the rigid girder-like construction of the two-section table, bottom left. One half of the table is adjustable in elevation with respect to its stationary complement on machined dovetail guides. A handwheel-operated elevating screw adjusts the table level for cut depths as indicated by a linear scale at the dovetail joint. Adjusting and locking screws are provided on one side of the female guides for aligning and locking the table setting.

High-speed steel blades of the cutter head, below, have built-in clamping screws and jackscrew elevating plugs are provided to facilitate relocating the cutter blades after resharpening. The ball bearing head may be V-belt driven, or direct driven from a flange-mounted motor, left.

A protractor scale on the welded triangular-section side guide indicates angular adjustment for beveling edges. Three mounting pads on the feet of the stationary table members have hold-down bolt holes through their centers. Distortion of the table by hold-down forces is thus avoided, even though the jointer may be mounted on an irregular bench surface instead of the fabricated base shown.

BOLT DESIGN FOR REPEATED LOADING

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A new approach to strength calculations for bolts subjected to periodically changing loads

CONVENTIONAL methods for calculating the strength of bolts are based upon the assumption that the bolted members are rigid bodies. These methods are satisfactory for assemblies under steady load conditions. However, they become unsatisfactory if the load acting on the bolted assemblies periodically changes.

This article will develop a rational method for calculating the strength of bolts for repeated loading conditions based on actual forces acting on the bolt, the range ratio of these forces, stress concentration, local stresses, and fatigue properties. Also included is a study of how size and form of individual bolt parts influence the reliability of bolts.

Recently a number of articles have been published[1, 6, 7, 40] suggesting methods of calculation which consider some of the factors influencing strength of bolts under repeated loading. The following analysis presents another method which introduces effects of these factors.

Actual Loads Acting on the Bolt: The symbols used throughout are

c_b = Tensile stiffness constant of the bolt, pounds per inch
c_c = Compressive stiffness constant of the connected parts, pounds per inch
F = External load on a single bolt, pounds
P = Tension load due to initial tightening of the nut, pounds
F_e = Effective additional tension force acting on the bolt due to the external force F, pounds
P_c = Critical tightening load, pounds.

This critical tightening load, P_c, is the smallest magnitude of the load P which will prevent the connected parts from being separated from each other when the external load F is applied.

The tightened bolted assembly can be represented schematically as a system which consists of two springs, *Fig.* 1. One of these springs represents the bolt and is extended; the other spring represents the connected parts and is compressed. Due to initial tightening by the force P, the bolt extends elastically by the amount δ_b, whereas the surrounding hollow body is compressed by the amount δ_c, *Fig.* 2. The tan-

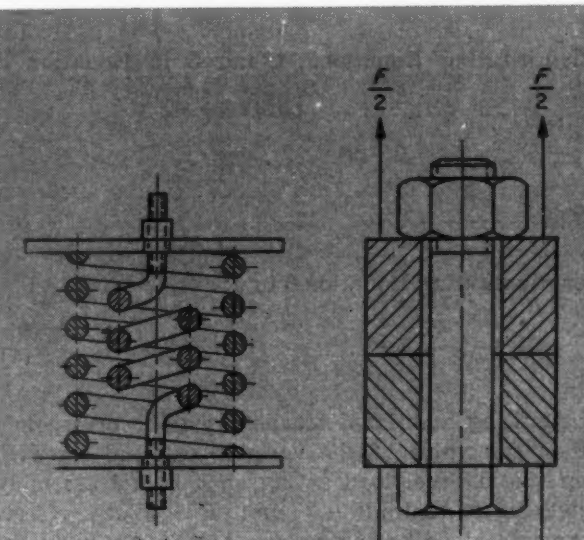

Fig. 1—Spring analogy for a bolted assembly

[1] References are tabulated at end of article.

gent of the angle θ is the ratio between the force and the corresponding bolt extension

$$\tan \theta = \frac{P}{\delta_b} \qquad (1)$$

This is equal to the tensile stiffness constant of the bolt, c_b. Similarly, $\tan \phi$ is the ratio between the force P and the corresponding compression of the connected parts.

$$\tan \phi = \frac{P}{\delta_c} \qquad (2)$$

Thus, $\tan \phi$ is equal to the compression stiffnes constant of the connected parts, c_c.

The triangle *ABC* can be constructed. By adding the external force F to the system, the bolt experiences an additional extension $\Delta\delta_b$ and the final compression of the spring representing the connected parts becomes smaller by the amount $\Delta\delta_b$. Therefore, the bolt force P increases by amount F_v to the amount F_{max}. Simultaneously, the load on the compressed parts decreases by the amount P_3 to the magnitude of P_2. The sum of the forces F_v and P_3 must be equal to the external force F for the condition of equilibrium, namely

$$F_v + P_3 = F \qquad (3)$$

Therefore, the polygon *ABDEC* takes a fixed form. With the help of the diagram, *Fig.* 2, it is possible to determine how the magnitudes of P_c and F_v depend upon the force F and the stiffness constants c_b and c_c.[43]

From the triangle *BDF*

$$F_v = (\Delta \delta_b) \tan \theta = (\Delta \delta_b) c_b \qquad (4)$$

Similarly, from triangle *BFE* it follows that

$$P_3 = (\Delta \delta_b) \tan \phi = (\Delta \delta_b) c_c \qquad (5)$$

By substituting Equations 4 and 5 in Equation 3

$$F = P_3 + F_v = (\Delta \delta_b)(c_c + c_b)$$

or

$$\Delta \delta_b = \frac{F}{c_b + c_c} \qquad (6)$$

After substituting Equation 6 in 4:

$$F_v = (\Delta \delta_b) c_b = F \frac{c_b}{c_b + c_c} \qquad (7)$$

The resulting additional tensile force F_v corresponds to the peak value of the variable external force F. When the external load acting on a bolted assembly changes in the limits $F = F_1$ up to $F = F_2$, the effective variable load acting on the bolt has a range of values from F_{v1} to F_{v2} as shown by Equation 7. Also the magnitude of P_c could be found. According to the definition, the critical load P_c, is of such a magnitude that the connected parts are compressed with a minimum force approaching zero as the external load approaches its maximum value F. With the external loading F, $P_2 = 0$. From the diagram it follows that

$$P = P_2 + P_3$$

and with $P_2 = 0$, then $P_c = P_3$. From Equation 3

$$P_c = F - F_v \qquad (8)$$

By substituting Equation 7 in Equation 8:

$$P_c = F - F \frac{c_b}{c_b + c_c} = F \frac{c_c}{c_b + c_c} \qquad (9)$$

If the initial tightening load equals P_c, the force with which the members are pressed together is changed between the limits P_c and 0 during the reversal of the external load between 0 and F. In order to avoid an impact condition and to insure a leak-proof fit in the assembly (cylinder head, air chamber, etc.), it is customary to make the magnitude of P larger than P_c so that

$$P = kP_c \qquad (10)$$

in which k usually varies from 1.2 to 1.5, but can also be greater. The value of k depends on the type of the bolted assembly and on the accuracy with which the external force F and the stiffness constants, c_b and c_c, may be determined.

The dependence of P_c and F_v on the ratio of stiffness constants c_b and c_c is graphically represented in *Fig.* 3. We see that the stiffness ratio very greatly influences the additional variable force F_v. In strength calculations it is important to recognize not only the maximum effective force acting on the bolt but also the change of the force with respect to time, which also greatly influences the strength of the bolt. The type of loading is conveniently represented by the range ratio, *R*, of the limit values of the loading cycle; that is, the ratio between the lowest load value to the highest value, considering their signs:

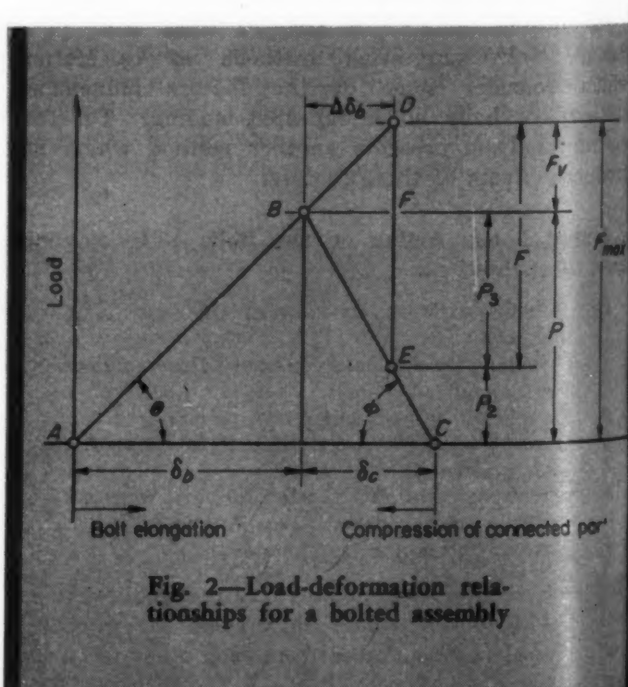

Fig. 2—Load-deformation relationships for a bolted assembly

$$R = \frac{F_{min}}{F_{max}} \qquad (11)$$

If the frequent case is taken in which the external load on the bolted assembly changes from 0 to F, and if the magnitude of the initial tightening force is taken equal to P_c, then:

$$F_{min} = P_c = F \frac{c_c}{c_b + c_c}$$

and

$$F_{max} = P_c + F_v = F \left(\frac{c_c}{c_b + c_c} + \frac{c_b}{c_b + c_c} \right) = F$$

and the range ratio of the tension force acting on the bolt is, therefore,

$$R = \frac{F \dfrac{c_c}{c_b + c_c}}{F} = \frac{c_c}{c_b + c_c}$$

In practice the tightening load P must be somewhat greater than P_c. Therefore, the type of bolt loading is somewhat different and, consequently, the range ratio is somewhat higher than the value $c_c/(c_b+c_c)$. However, the previous example shows the manner in which the stiffness constants c_b and c_c influence the type of bolt loading.

Thus, from the previous discussion and consideration of *Fig.* 2, the equations for the general case are

$$F_{max} = P + F_v = kF \frac{c_c}{c_b + c_c} + F \frac{c_b}{c_b + c_c} \qquad (12)$$

and also

$$P_c + F_v = F \qquad (13)$$

Of the two forces, P_c and F_v, the force F_v is more critical for two reasons. First, strength of materials is lower for variable loading than it is for steady loading; and secondly, stress concentration affects the strength considerably when the load is variable. Thus, in order to increase the reliability of the bolt, the force F_v should be decreased at the expense of increasing the force P_c. This is clearly seen by referring to *Fig.* 3. Hence, the rule for the design of bolt assemblies subjected to repeated loading can be stated, "The stiffness of the bolted members must be as high as possible and the stiffness of the bolts as low as possible."[1]

Determination of Twisting Moment: With the tightening of the nut, a twisting moment arises due to friction between the nut and bolt threads. This moment may be determined from

$$M = Pr_m \frac{\lambda + \gamma}{\cos \alpha}$$

where r_m = mean radius of the thread, inches; λ = helix angle of thread at the mean diameter; γ = angle of friction; α = half angle of thread, measured on the plane through the axis.

In practice, it is more convenient to use

$$M = \beta PD \qquad (14)$$

where β = coefficient, determined experimentally; D = nominal diameter of the bolt, inches.

Coefficient β depends upon the magnitude of the angle of friction as well as upon the type of thread. Apparently the type of thread has little influence on the twisting moment, M. Value of the angle of friction, however, varies significantly. Results of numerous experiments show that coefficient β depends on: (1) materials of the bolt and nut, (2) type of heat treatment, (3) method of forming the threads, (4) condition of thread surface due to wear, (5) kind of plating, if any, (6) kind of lubrication, etc. Be-

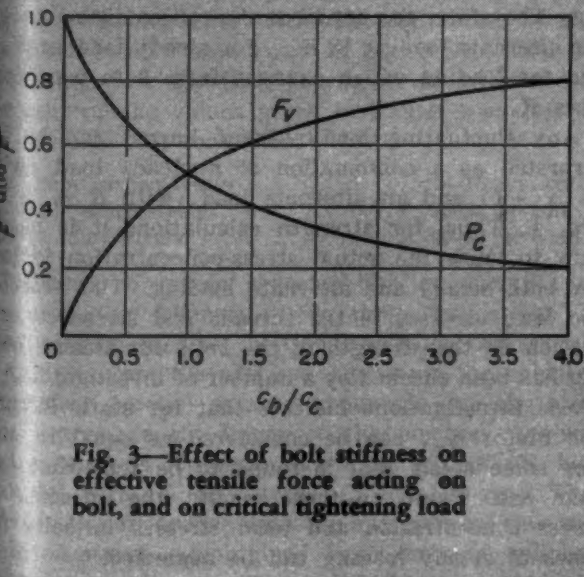

Fig. 3—Effect of bolt stiffness on effective tensile force acting on bolt, and on critical tightening load

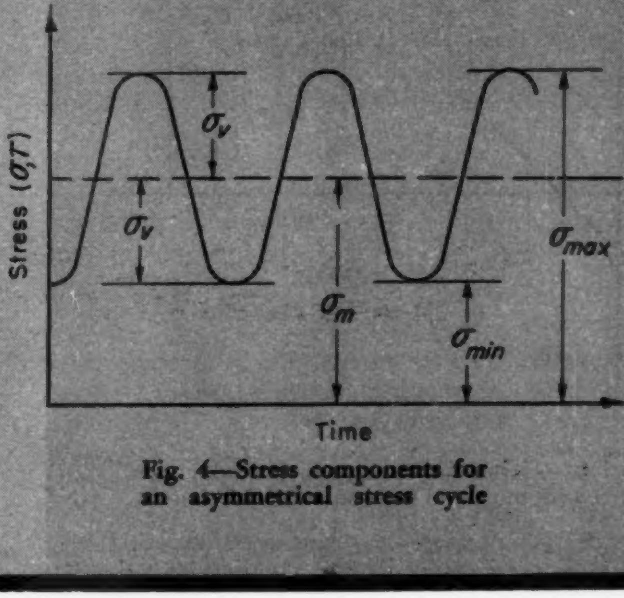

Fig. 4—Stress components for an asymmetrical stress cycle

cause of these many variables, the value to be used for M in strength calculation, when the bolts are highly stressed, should be based upon experimental data for the case under consideration.

It must be remembered that torque on the wrench differs from the moment, M, which affects strength of the bolts. Torque on the wrench overcomes friction resistances both between the nut and bolt, and between the nut and supporting surface, while the moment M, which loads the bolt, depends upon friction in the threads only.*

If specific experimental data are unavailable, average values of coefficient β for conventional bolt threads which can be used in Equation 14 are:[2-7]

Well-lubricated, smooth surfaces	0.10
Unlubricated, smooth surfaces	0.12
Well-lubricated, surfaces not smooth	0.13-0.15
Unlubricated, surfaces not smooth	0.18-0.20

Unlubricated refers to practically dry thread surfaces (i.e., surfaces that have not been intentionally lubricated but nevertheless could contain a small residue of oil). Well-lubricated refers to oil lubrication. Since repeated tightening of bolts eventually wears out the thread surfaces, values of β under these circumstances must be based on thread surfaces not considered smooth. Values of β given are for steel bolts with the threads cut—not rolled or ground—and without plating.

If bolts are highly stressed, it is desirable to relieve the bolt of torque. If this is done, the twisting moment, M, does not need to be determined for strength calculation. This condition is considered later.

* In general, experiments show, the relationship between torque on the wrench and initial tightening of the nut, P, is very inconsistent and, therefore, results of calculating the force P by torque on the wrench are unreliable. Apparently, the only reliable method at present to determine initial tightening of the nut is by actual measurement of bolt elongation.

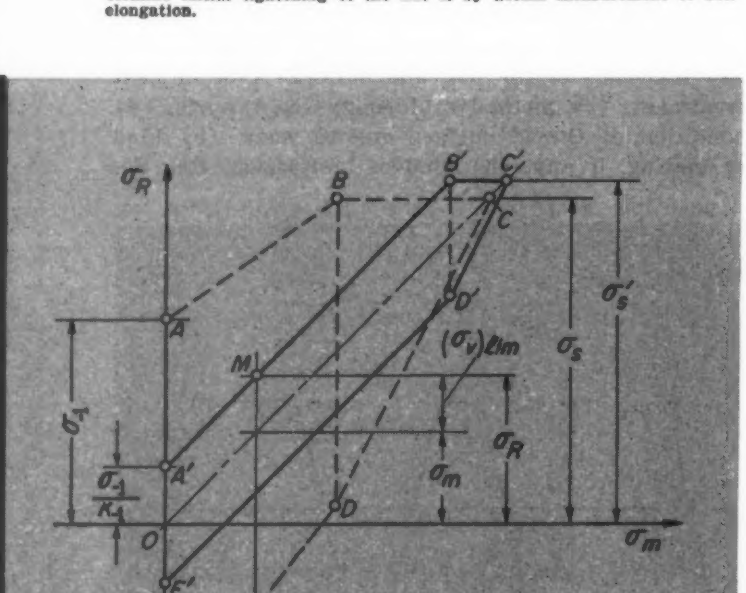

Fig. 5 — Goodman diagram for threaded bolts with nuts, and for notch-free specimens

Stress Concentration and Local Stresses: On the threaded portion of the bolt which is inside the nut, the following loads act simultaneously: (1) the bolt is stretched by the initial load P and by the additional variable force F_v; (2) the root cross-section is twisted by the moment, which arises with the initial tightening of the nut; (3) the threads experience bending due to the pressure of the nut; (4) due to the pressure of the nut, shearing stresses arise in the screw threads; (5) compression appears between the thread surfaces.

Distribution of the axial load among the individual threads is not uniform; thus, the distribution of bending, shearing, and compressive stresses among the individual threads is also not uniform. The nonuniformity of distribution of load among the threads arises due to elastic deformation of the bolt and nut under load. Therefore, this nonuniformity appears even in cases of ideal accuracy in thread production. By using a nut of a special shape, it is possible to decrease the nonuniformity of force distribution among the individual threads and to increase—considerably—the endurance strength of the bolt.

Because of the great complexity of the stress conditions at the critical cross-section of the bolt, the analytical determination of the maximum stress at these points is hardly possible. Therefore, the actual or effective stress concentration factor, K, must be applied.

This factor represents the ratio of the strength of the polished specimen containing no discontinuities to the strength of the given member of the same material and having the same critical cross-sectional area. In determining the actual stress-concentration factor, the same loading range ratio, R, must be applied to both the polished specimen and the member.

The subscript with K indicates the magnitude and the sign of the range ratio of the corresponding loading. Therefore, the actual stress-concentration factor for alternate loading is K_{-1}, for steady loading K_{+1}, and for loading which changes from 0 to maximum it is K_0.

Any fluctuating load without impact can be interpreted as a combination of a steady load (with $R = +1$) and an alternate load (with $R = -1$), *Fig.* 4. Thus, for strength calculations it is necessary to know the actual stress-concentration factors for both steady and alternate loading. The effect of the local stresses in the threads and stress concentration on the strength of the bolts for steady loading has been checked by a number of investigators.[8-14] These investigations indicate that for static loading the factor K_{+1} can be considered as equal to one. For some steels K_{+1} is found to be somewhat less than one. Based on these results, the influence of stress concentration and local stresses in bolts for cases of steady loading can be neglected.

Such is not the case for alternate loading, where

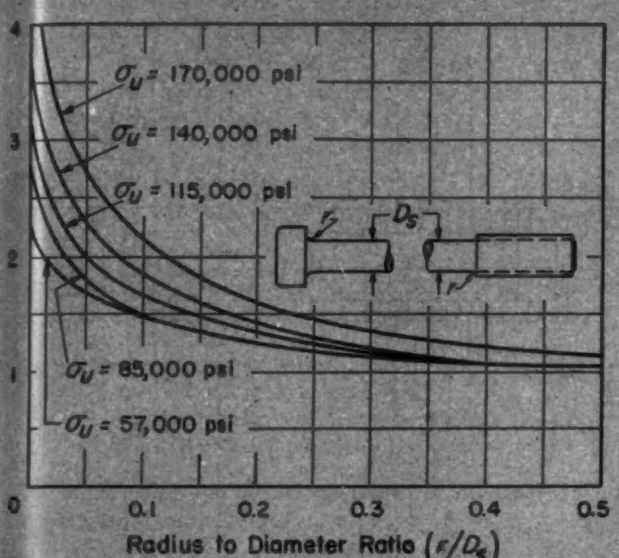

Fig. 6 — Stress-concentration factors for steel bolts at a section containing fillets

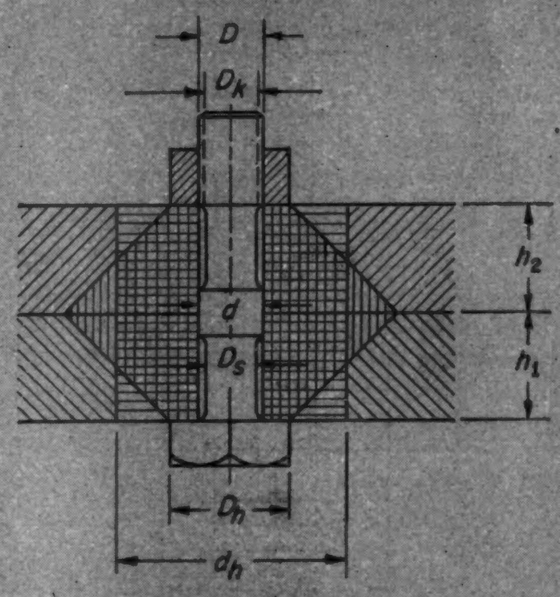

Fig. 7—Schematic representation of flanges undergoing compressive deformation

stress concentration and local stresses greatly influence the strength of bolts. The effect of these factors on the strength of the bolts for variable loading has been investigated by a number of investigators.[9, 12, 15-25] In some of these experiments the bolts were investigated under conditions similar to those existing in actual bolted assemblies. Available data on actual stress-concentration factors in threads are still not quite sufficient for reliable determination of their magnitudes. However, on the basis of data published it is possible to accept for the present the values shown in TABLE 1 for K_{-1}

Values in TABLE 1 refer to bolts with the threads machined by cutting. When the threads are not heat-treated after forming and are either formed by rolling or by rolling after initial cutting, values of K_{-1} are lower than those given in the table. This decrease due to the rolling process could be stated in per cent of the values given in TABLE 1 as follows for various bolt materials:[9, 23-30]

Medium carbon steels	14-18 per cent
Ductile alloy steels	20-24 per cent
Hard alloy steels	30-40 per cent

Influence of stress concentration on strength of the bolts for different stress range ratios can be clearly represented by the Goodman's diagram, *Fig*. 5. In this diagram the abscissa represents the value of the steady (mean) stress components and the ordinate represents the limits of the nominal stresses. The point A is the endurance limit σ_{-1}, of the bolt material, with a steady stress, σ_m, equal to zero. This point is the endurance limit for symmetrical reversed cycle of stress, as obtained from tests of polished notch-free specimens. The point A' corresponds to the endurance limit of the threaded specimens, with nuts. This ordinate is equal to σ_{-1}/K_{-1}. Point C is the limit value for steady loading. It corresponds to the yield point of the material, σ_s. Point C' is the yield point σ'_s, as obtained from the test of threaded specimens with nuts. Line $ABCDEA$ represents the endurance limits for notch-free specimens for different range ratio of loading cycles. The line $A'B'C'D'E'A'$ has the same meaning but for specimens with threads and nut. The diagram shows not only the influence of the notch but the combined influence of all factors which lower the design strength of the bolt and nut assembly. Reliable experiments by Pomp and Hempel[12] show that the line $A'B'$ could be taken as a line parallel to the straight line OC (zero to σ_s) with sufficient accuracy for practical calculations. That is, for repeated bolt loading, the limiting amplitude of the alternate stress components $(\sigma_v)_{lim}$ at point M, for example, does not depend upon the value of the steady stress component. With this form of endurance diagram, the value of the stress limit, σ_R, for an asymmetrical loading cycle can be expressed by the formula:

$$\sigma_R = \sigma_m + \frac{\sigma_{-1}}{K_{-1}} \quad (15)$$

Nonuniformity of the stress distribution occurs at the junction between the shank and the head and at other places where the diameters change. The magnitude of the effective stress-concentration factors can be taken from the diagram, *Fig*. 6.[31, 32]

Stiffness of Bolts and Connected Parts: Stiffness constant of the bolt can be determined according

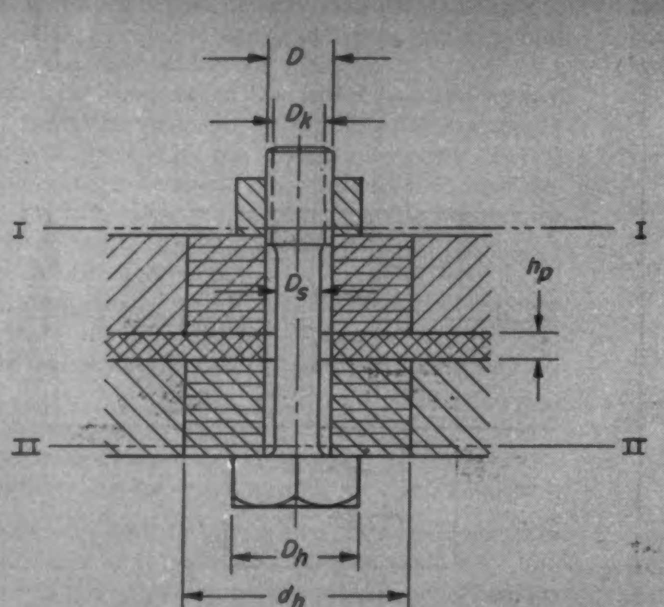

Fig. 8 — Flanges, with gasket or plate between, undergoing compressive deformation

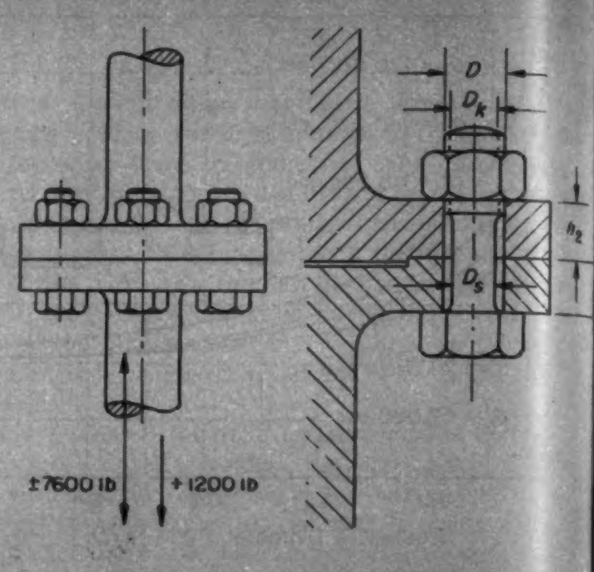

Fig. 9 — Typical assembly loaded with steady and alternating forces

to the formula:

$$\frac{1}{c_b} = \frac{1}{c_1} + \frac{1}{c_2} + \frac{1}{c_3} + \ldots\ldots + \frac{1}{c_n} = \sum \frac{1}{c_i} \qquad (16)$$

where c_b = stiffness constant of the bolt; $c_1, c_2 \ldots . c_n$ are the stiffness constants of the individual bolt portions.

$$c_i = \frac{E_b\, A_i}{L_i} \qquad (17)$$

where E_b = modulus of elasticity of bolt material, psi; A_i = cross-section area of the corresponding bolt portion, square inches; L_i = length of this portion, inches. Stiffness of the connected parts, as they are compressed, can be defined analytically when the forms of the parts are relatively simple.

The assembly of two flat plates bound together with bolts is an example, *Fig.* 7. In this case the stiffness constant of the connected parts is

$$c_c = \frac{E_c\, A_c}{h_1 + h_2} \qquad (18)$$

where E_c = modulus of elasticity of the flanging materials in compression, psi; $h_1 + h_2$ = flange thickness, inches; A_c = effective cross-section area of the flange portions which are undergoing deformation, square inches.

These flange cross-sections can be represented by compression cones, as shown in *Fig.* 7, which cut the bearing surfaces under the nut and the head at an angle of about 45 degrees. Elasticity of a double cone can be determined approximately by replacing the cones with a hollow cylinder using the mean cone diameter as the outer cylinder diameter and the same inside diameter, *d*, as the cone.[19, 33]

Thus, the effective cross-section area is

$$A_c = \frac{\pi}{4}\,(d_h^2 - d^2) \qquad (19)$$

If the outside diameter of the bearing surface of the nut is set equal to the outside diameter of the bearing surface of the head and is designated as D_h the diameter of the compression cylinder, d_h, becomes

$$d_h = D_h + \frac{h_1 + h_2}{2} \qquad (20)$$

If a thin plate is placed between the flanges, as shown in *Fig.* 8, the general stiffness constant is determined according to

$$\frac{1}{c_c} = \frac{1}{c_f} + \frac{1}{c_p} \qquad (21)$$

where c_f = flange stiffness constant in compression obtained as c_c from Equation 18; and c_p = stiffness constant of the plate, which is determined by

$$c_p = \frac{E_p\, A_c}{h_p}$$

where E_p = modulus of elasticity in compression of the plate material, psi; h_p = thickness of the plate, inches.

As was previously mentioned, stiffness of the bolted members may be determined analytically with sufficient accuracy for the case of a relatively simple

form. When the assembly of the machine members is complex in form (e.g., in the bolted assembly of a connecting rod), the stiffness constant, c_a, must be determined experimentally. Experiments of this kind have been carried out in order to establish the relation between the stiffness constants, c_b and c_a, on some bolted assemblies.[19, 34, 35, 36]

Determination of Safety Factors: Determining safety factor of the bolt involves the following stresses:

- σ_m = Nominal normal stress in the critical bolt cross-section due to the steady components of the load, pounds.
- σ_v = Nominal normal stress due to the alternate component of the load, pounds.
- τ = Nominal shearing stress due to the steady twisting moment, pounds.

If the nominal normal stress in the cross-section under consideration changes from σ_{min} to σ_{max}, the steady stress component, *Fig.* 4, is equal to

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \qquad (23)$$

and the alternate stress component is equal to

$$\sigma_v = \frac{\sigma_{max} - \sigma_{min}}{2} \qquad (24)$$

If shaft diameter is equal to thread diameter, the critical cross-section is located where the threaded part of the bolt enters the nut. This is confirmed by experiments and observation of the failure of bolts in bolted assemblies working under repeated loading. However, if the bolt shank diameter is smaller than the minor thread diameter, the shank diameter must also be calculated at the place where the greatest stress concentration factor is to be expected, *Fig.* 8.

Bolts are not always loaded with the torsional moment. In particular cases where the bolts are highly stressed (bolts in connecting rods of internal combustion engines), it is desirable to turn the nut back a small angle after it has been tightened. This must be done in such a way that the bolt will be relieved of the torque without changing the magnitude of the initial tension force along the axis. Best results can be obtained when the torque is released with the nut already locked on the bolt.

In this case, when only the tension forces act on the bolt, the following expression may be used for the determination of the factor of safety:

$$N = \frac{\sigma_R}{\sigma_{max}} \qquad (25)$$

This is the ratio of the nominal stress limit (endurance limit for the given stress range ratio) to the maximum nominal stress at the cross-section under consideration, in which σ_R is obtained from Equation 15 and, from *Fig.* 4,

$$\sigma_{max} = \sigma_m + \sigma_v \qquad (26)$$

By substituting Equations 15 and 26 in Equation 25, the safety factor is

$$N_1 = \frac{\sigma_R}{\sigma_{max}} = \frac{\sigma_m + \dfrac{\sigma_{-1}}{K_{-1}}}{\sigma_m + \sigma_v} \qquad (27)$$

It must be mentioned that Equations 15 and 27 are only true when

$$\sigma_{max} = \sigma_m + \sigma_v < \sigma_s$$

$$\sigma_m + \frac{\sigma_{-1}}{K_{-1}} < \sigma_s$$

which must be the case for bolted assemblies.

Safety factor of bolts loaded with both tensile forces and torsional moments may be calculated from

$$\frac{1}{N_2^2} = \left(\frac{\sigma_{max}}{\sigma_R}\right)^2 + \left(\frac{\tau}{\tau_s}\right)^2 = \left(\frac{\sigma_m + \sigma_v}{\sigma_m + \dfrac{\sigma_{-1}}{K_{-1}}}\right)^2 + \left(\frac{\tau}{\tau_s}\right)^2 \qquad (28)$$

where τ = the nominal shearing stress in the given cross-section, psi; τ_s = torsional shearing yield strength of the bolt material, psi. This expression is based on the elliptical form of the limiting curve for normal and tangential stresses acting simultaneously. It seems that this form of limiting curve agree satisfactorily with the available experimental data for ductile metals as used for bolts.[32]

Equations 27 and 28 for calculating the strength of bolted assemblies were suggested by the author in an article published in 1940.[27]

Another method for calculating the strength of bolts suggested by Doughtie and Carter[6] was published in 1950.[7, 40] These authors suggested a graphical method for determining the limiting nominal stress in bolts for different ranges of stress using a simplified steady stress-alternating stress diagram for steel. If this limiting nominal stress is expressed analytically, it takes the form

$$\sigma_R = \sigma_m + \frac{\sigma_{-1}}{K_{-1}}\left(1 - \frac{\sigma_m}{\sigma_s}\right)$$

This form has not been confirmed experimentally for bolts. Experiments[12] show that the magnitude of the limiting alternate stress component does not depend upon the magnitude of the mean stress component σ_m. Thus, it seems that Equations 15 and 27 will give more reliable results. Doughtie and Carter also suggested using the stress concentration factor

Table 1—Stress Concentration Factors for Alternately Loaded Bolts*

Thread Form	Medium-carbon Steels	Heat-treated Alloy Steels
Whitworth thread and unified thread with rounded root	3.2-3.8	5.4-6.0
Metrical standard DIN and OST	4.4-5.0	5.6-6.4
Sellers (American National form with flat root)	5.0-5.8	6.4-7.2

* Lower values for softer and more ductile materials.

found by Moore and Henwood[15] as a means for determining the endurance limit for complete stress reversals. It may be noticed that these factors are found for the load on the specimens which varies from nearly zero to the maximum value. Therefore, use of these factors for determining the endurance limits for completely reversed loading cycle could cause considerable error in the results.

Flange Joint Example: As a demonstration of the suggested method, determine the safety factor for the flange connection shown in *Fig.* 9. The joint is loaded with a steady load, $W = 1200$ pounds, and with an alternating load, $Q = \pm 7600$ pounds. Number of bolts in the joint is four. Flange thickness is, $h_1 = h_2 =$ 0.8-inch. The bolts have UNC threads. Basic major diameter $D = 1$ inch; minor diameter of the screw thread $D_K = 0.846$-inch; root area of thread $A_K =$ 0.562-square inch; shank diameter $D_s = 0.8$-inch; shank cross-section area, $A_s = 0.5027$-square inch.

Bolt stiffness constant is, from Equation 17,

$$c_b = c_i = \frac{E_b A_s}{L_i} = \frac{(30 \times 10^6)(0.5027)}{1.6}$$
$$= 9.43 \times 10^6 \text{ pounds per inch.}$$

Flange stiffness constant is determined from Equations 18, 19 and 20. Outside diameter of the bearing surface of the nut, D_h, is given as 1.5 inches. From Equation 20, therefore, d_h is 2.3 inches and from Equation 19, A_c becomes 3.29 square inches. By using a value of 30×10^6 for E_c in Equation 18, the flange stiffness constant, c_c, works out to 61.7×10^6 psi.

Since $F = (Q + W)/4$, critical tightening force can be determined from Equation 9 to be 1910 pounds. By taking the magnitude of the factor k equal to 1.3, from Equation 10 the magnitude of the initial tightening force, P, is approximately 2480 pounds.

The additional alternate force, F_v, imposed by the alternating load can be found from Equation 7, in which F is considered equal to $Q/4$. The force F_v thus is 252 pounds.

Similarly, the additional steady force is

$$F' = \frac{W}{4}\left(\frac{c_b}{c_b + c_c}\right) = 39.75 \sim 40 \text{ pounds.}$$

Torsional Moment: If the coefficient $\beta = 0.15$ is used, Equation 14 gives moment, $M = 372$ lb-in. Smallest nominal stress due to the tension in the bolt cross-section with the diameter D_K is

$$\sigma_{min} = \frac{P + F' - F_v}{A_K} = 4035 \text{ psi.}$$

Greatest nominal stress in the same cross-section is

$$\sigma_{max} = \frac{P + F' + F_v}{A_K} = 4932 \text{ psi.}$$

Nominal steady stress component σ_m from Equation 23 is, therefore, 4484 psi, and nominal alternating stress component, σ_v, from Equation 24 is 448.5 psi. Nominal shearing stress in this cross-section due to the torsional moment is:

$$\tau = \frac{M}{Z_0} = \frac{16\,M}{\pi\,D_K^3} = 3130 \text{ psi}$$

where Z_o = polar section modulus, inches^3.

Material of the bolts is a carbon steel having the properties: $\sigma_u = 78{,}500$ psi; $\sigma_s = 44{,}000$ psi; $\tau_s =$ 27,000 psi; $\sigma_{-1} = 26{,}000$ psi. According to Equation 15, the fatigue strength with the given range ratio is

$$\sigma_R = \sigma_m + \frac{\sigma_{-1}}{K_{-1}} = 4484 + \frac{26{,}000}{3.5} = 11{,}915 \text{ pounds}$$

where $K_{-1} = 3.5$ (Table 1) is the mean value for the actual stress-concentration factor for this form of threads on a carbon-steel bolt. By substituting these values in Equation 28, $1/N_2^2 \approx 0.184$, from which the safety factor $N_2 \approx 2.33$.

By neglecting the shearing stress due to the torsional moment, Equation 27 could be used for determining the safety factor. In this case $N_1 = 2.42$. The results show that the torsional moment has little influence on the strength of these bolts.

Connecting Rod Example: Check the fatigue strength for the bolted assembly of a connecting rod of a two-cycle Diesel engine. Number of bolts in the assembly is four. UNF threads are used, with a major diameter, $D = 1.0$ inch and the minor diameter of the screw thread $D_K = 0.8978$-inch. Root area A_K

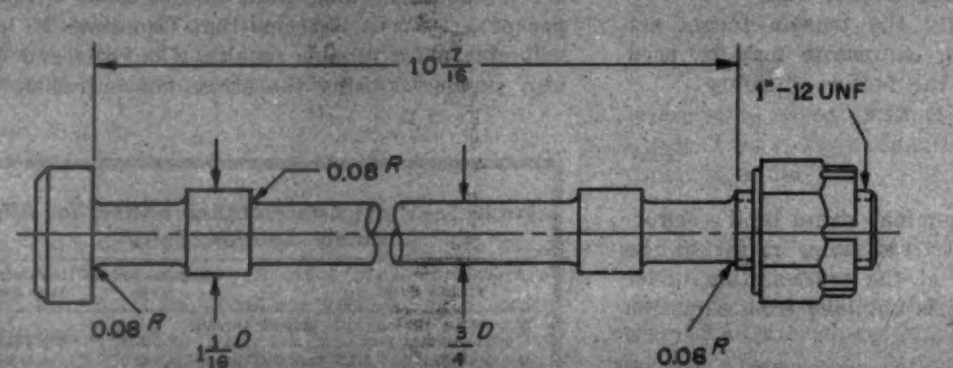

Fig. 10—Diesel-engine connecting-rod bolt

= 0.633-square inches; the fillets of the shank have a radius $r = 0.08$-inch. Other dimensions are shown in *Fig.* 10. The bolted assembly is loaded with an inertia force $F_m = 32{,}300$ pounds, which changes from zero to its maximum value during each revolution of the engine. The bolts in assembly are unloaded from the torsional moment.

External load which comes on a single bolt is

$$F = \frac{32{,}300}{4} = 8075 \text{ psi}$$

Based on published experimental data,[19] we may consider the ratio between the bolt stiffness and the stiffness of the corresponding parts of the connecting rod as 1:1.5, that $c_c = 1.5\ c_b$. Thus, the value of the critical tightening force is

$$P = F\frac{c_c}{c_b + c_c} = 8075\frac{1.5}{1.0 + 1.5} = 4845 \text{ pounds.}$$

If $k = 1.4$, then $P = 1.4\ P_c = 6780$ pounds. The variable tension force due to the external load is

$$F_v = F\frac{c_b}{c_b + c_c} = 8075\frac{1.0}{1.0 + 1.5} = 3230 \text{ pounds.}$$

Nominal stresses at the critical cross-section at the thread are

$$\sigma_{min} = \frac{P}{A_K} = \frac{6780}{0.633} = 10{,}700 \text{ psi;}$$

$$\sigma_{max} = \frac{P + F_v}{A_K} = \frac{6780 + 3230}{0.633} = 15{,}800 \text{ psi;}$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = 13{,}250 \text{ psi;}$$

$$\sigma_v = \frac{\sigma_{max} - \sigma_{min}}{2} = 2550 \text{ psi.}$$

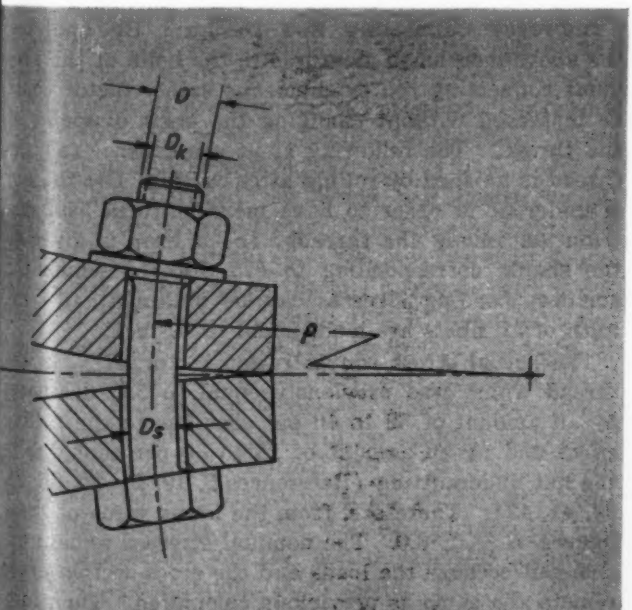

Fig. 11—Bending deformation of bolt due to deformation of connected parts

The bolts are machined from chrome-nickel heat-treated steel with the properties: $\sigma_u = 140{,}000$ psi; $\sigma_s = 114{,}000$ psi; $\sigma_{-1} = 47{,}000$ psi. Mean value of the actual stress concentration factor according to TABLE 1 is $K_{-1} = 5.7$.

Thus, the fatigue strength for the given range ratio, σ_R, from Equation 15 is 21,500 psi and the safety factor, N_1, approximately equals 1.36. The effect of bending stresses is not considered in this analysis. This effect may be rather significant for bolts in a connecting rod which has relatively flexible bolted members. Therefore, the safety factor 1.36 is probably too small for this kind of bolted assembly.

SAFETY FACTOR AT SHANK CROSS-SECTION: The critical shank cross-section is located at the fillet between the shank and the head. The fillet radius is equal to 0.08-inch. The shank diameter is $D_s = 0.75$-inch. Cross-section area is $A_s = 0.4418$-square inches. Then $\sigma_{min} = 15{,}350$ psi; $\sigma_{max} = 22{,}670$ psi; $\sigma_m = 19{,}010$ psi; and $\sigma_v = 3660$ psi. According to *Fig.* 6, the actual stress concentration factor for the steel used is equal to $K_{-1} = 1.9$ to 2.0. We take $K_{-1} = 2.0$. The fatigue strength of the shank for the given stress range ratio is $\sigma_R = 42{,}510$ psi, and the safety factor is $N_s = 1.87$. This shows that the critical section is at the threaded portion of the bolt.

Improving Bolt Performance: Reliability of a bolted assembly can be increased considerably without changing the basic diameter of the bolt. This could be accomplished by decreasing four factors: (a) effective stress-concentration factor in the threads; (b) longitudinal stiffness of the bolt; (c) magnitude of K_{-1} in the critical shank cross-section and; (d) bending stresses in the bolt.

At the threads K_{-1} can be decreased by rolling the threads, or by using a rolling process after cutting. Another method is to use a special nut shape which results in a more uniform load distribution among the individual threads than in an ordinary nut.

Longitudinal bolt stiffness could be decreased by decreasing the shank cross-sectional area. However, this method is limited because of the simultaneous increase of stress in the shank.

The magnitude of K_{-1} at the critical shank cross-section could be decreased in some cases by increasing the fillet radii at the junction between the shank and head and at other places where the diameters change.

BENDING STRESSES: Repeated bending stresses occur in the bolts due to deformation of the bolted members under external load. Usually the bending stiffness of the bolted members is much higher than the bending stiffness of the bolts. Therefore, the magnitude of the bending deformation of the bolts depends primarily upon the stiffness of the bolted members. It is desirable to reduce the bending moment that acts on the critical bolt cross-section as much as possible. Generally, this critical section is located at the threaded portion of the bolt. The bending moment acting on the shank depends upon the flexibility of the shank. If the radius of curvature, ρ, which depends mostly upon the flexibility of bolted

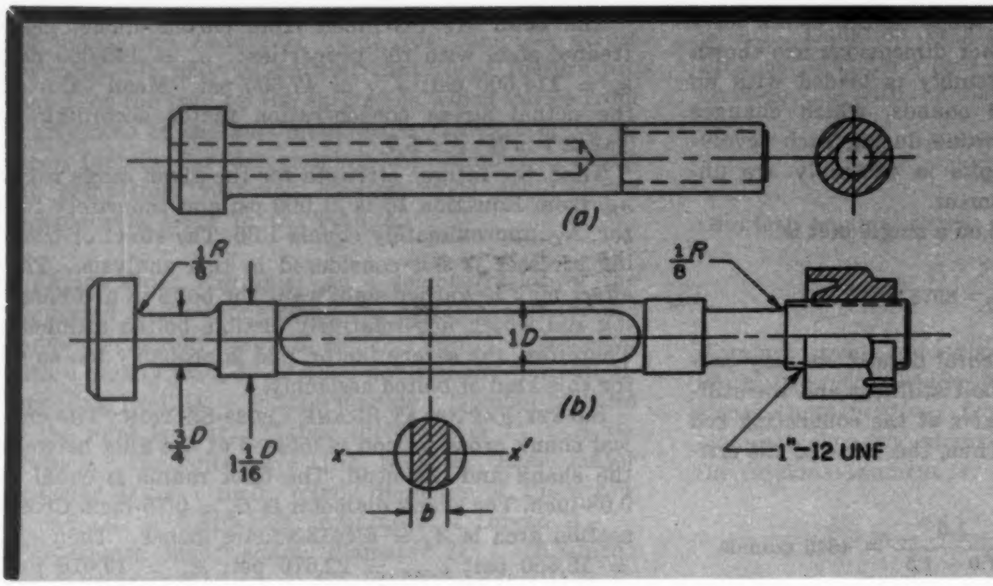

Fig. 12—Bolt design, *a*, using a drilled hole as a means of increasing flexibility, and improved design, *b*, for bolt shown in *Fig.* 10

members, is given, this bending moment is equal to

$$M_b = \frac{E_b I}{\rho} = \frac{E_b \pi}{64 \rho} D_s^4 \quad \text{.......... (29)}$$

where E_b = modulus of elasticity of the bolt material, psi; I = moment of inertia of the shank cross-section, inches[4]; ρ = radius of curvature of the bolt axis, inches; D_s = shank diameter, inches (see *Fig.* 11).

The same moment, M_b, acts on the critical cross-section of the threaded portion of the bolt. Therefore, the nominal bending stress at the threaded portion of the bolt is equal to

$$\sigma_b = \frac{M_b}{Z_b} = \frac{\dfrac{\pi E_b D_s^4}{64\rho}}{\dfrac{\pi D_K^3}{32}} = \frac{E_b D_s^4}{2\rho D_K^3} = c_4 D_s^4 \quad \text{..... (30)}$$

where D_K = minor diameter of the screw thread, inches; Z_b = section modulus of the threaded portion of the bolt, inches[3]. Thus, by decreasing the shank diameter, the magnitude of the bending stress at the critical bolt cross-section decreases very rapidly. Since the bending moment and, consequently, the stress in the critical cross-section depends upon the moment of inertia of the shank cross-section, it is highly desirable to decrease, as far as possible, the moment of inertia of the shank. The actual radius of curvature, ρ, hardly can be calculated in most cases. Therefore, nominal bending stresses cannot be taken into consideration when determining the safety factor, although they influence rather significantly the strength of some bolted assemblies.[1, 36, 37] However, these bending stresses must be kept in mind in design.

From this standpoint, the design shown in *Fig.* 12*a* is very unsatisfactory. In this rather widely used design, the cross-sectional area is decreased by drilling a hole in the shank along its axis. By this means only the longitudinal stiffness can be decreased. Bending stresses in the shank remain the same as those in the solid shank having the same outside diameter. Bending moment and bending stresses at the critical cross-section (in threaded bolt portion) are only insignificantly smaller, since the moment of inertia of the ring at the shank cross-section differs very little from the moment of inertia of the whole cross-section with the same diameter.

Three different cross-sections with the same area are represented in *Fig.* 13. The cross-section at the right, seemingly, solves the problem of decreasing the bending stresses in the best manner, while the left cross-section does it in the poorest manner. The bolt with the shank cross-section represented at the right could be used with success if the repeated bending moment acts in one definite plane relative to the bolt, which is usually the case in machine parts. The presupposition is that the bolt bends in the plane X—X'. In the case of a circular flange joint the axes X—X' must coincide with the radii of the flange.

Improved Connecting Rod Design: By applying the above-mentioned measures to the bolts of the previous connecting rod example, the safety factor could be increased without changing the basic diameter of the thread. The following steps are taken: (a) the thread is finished by rolling after cutting; (b) the nut is undercut in order to have more uniform load distribution among the threads; (c) a form is given to the shank, corresponding to *Fig.* 12*b*, by which the smallest bending stresses can be obtained; (d) the radii of all fillets are made equal to ⅛-inch.

The actual stress concentration factor, K_{-1}, in the thread which was previously equal to 5.7, decreases by an amount of 30 to 40 per cent due to the rolling effect and by an amount of 25 to 30 per cent due to the nut undercutting (References 2, 19, 21, 26, 35, 38, 39, 41, 42). Therefore, from the lower value of these figures, $K_{-1} \approx 3.0$. The nominal stresses remain unchanged because the loads and the cross-section areas remain the same as previously calculated. Therefore, σ_R, reduces to 28,900 psi and the safety factor, N_1, becomes 1.83. This safety factor of 1.83 can be considered as satisfactory for this kind of assembly. The magnitude of the safety factor increases from 1.36 to

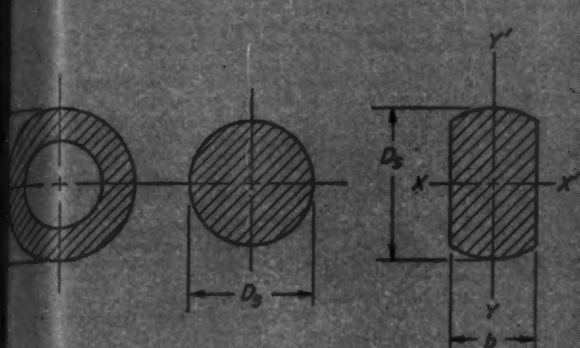

Fig. 13—Cross-sections of shanks having equal areas

1.83, which is a considerable increase in reliability of the design. Actually, the increase in reliability of the assembly is still higher than indicated because of the decrease of stresses due to repeated bending.

Safety Factor at Shank Cross-Section: The actual stress concentration factor with $R = 1/8$, according to *Fig.* 6, is $K_{-1} = 1.7$. The fatigue strength, σ_R, is 46,660 psi and $N_s = 2.03$. Thus, the critical section of the bolt is at the threaded portion as before.

Determining Optimum Shank Diameter: It was shown previously that the shank stiffness has considerable influence upon stresses at the threaded portion of the bolt where the critical cross-section is located. The favorable influence on the strength of assemblies under repeated loading obtained by increasing the bolt elasticity has been known for many years, and bolts with reduced stiffness have received general acknowledgment. The suggested method permits one to find the best relationship between the size of the threaded portion of the bolt and its shank diameter.

Usually there are two critical cross-sections where the safety factor of the bolt must be checked. These are section I—I at the thread and section II—II at the junction between the shank and head, *Fig.* 8. Sometimes the critical section II—II can be located at other points of the shank. We have already found that safety factor in the threads increases with a decrease of the shank stiffness. But, at the same time, when the shank cross-section decreases, stresses in the cross-section II—II increase, and the safety factor, N'', in this section decreases. Theoretically, highest reliability of the bolt will be reached when the safety factor at both critical section I—I and II—II are equal. An investigation can be conducted of the case which happens often in machine design—the external load acting on the bolted assembly changes from zero to maximum. Also assume that the bolt is unloaded from the torsional moment. At the cross-section I—I, *Fig.* 8,

$$\sigma_{min}' = k\left(\frac{F}{A'}\right)\left(\frac{c_e}{c_b + c_e}\right)$$

$$\sigma_v' = \left(\frac{F}{A'}\right)\left(\frac{c_b}{c_b + c_e}\right)$$

$$\sigma_{max}' = \sigma_{min}' + \sigma_v' = \frac{F}{A'}\left(k\frac{c_e}{c_b + c_c} + \frac{c_b}{c_b + c_e}\right)$$

$$\sigma_m' = \frac{\sigma_{max}' + \sigma_{min}'}{2} = \frac{F}{A'}\left[k\frac{c_e}{c_b + c_e} + \frac{c_b}{2(c_b + c_e)}\right]$$

$$\sigma_R' = \sigma_m' + \frac{\sigma_{-1}}{K_{-1}'}$$

$$= \frac{F}{A'}\left[k\frac{c_e}{c_b + c_e} + \frac{c_b}{2(c_b + c_e)}\right] + \frac{\sigma_{-1}}{K_{-1}'}$$

and the safety factor in cross-section I—I is

$$N' = \frac{\sigma_R'}{\sigma_{max}'} = \frac{k\dfrac{c_e}{c_b + c_e} + \dfrac{c_b}{2(c_b + c_e)} + \dfrac{A'}{F}\left(\dfrac{\sigma_{-1}}{K_{-1}'}\right)}{k\dfrac{c_e}{c_b + c_e} + \dfrac{c_b}{c_b + c_e}}$$

Similarly the safety factor in cross-section II—II of the shank is:

$$N'' = \frac{\sigma_R''}{\sigma_{max}''} = \frac{k\dfrac{c_e}{c_b + c_e} + \dfrac{c_b}{2(c_b + c_e)} + \dfrac{A''}{F}\left(\dfrac{\sigma_{-1}}{K_{-1}''}\right)}{k\dfrac{c_e}{c_b + c_e} + \dfrac{c_b}{c_b + c_e}}$$

In the previous equations A' and A'' are the cross-sectional areas in the section I—I and II—II, respectively, and K_{-1}' and K_{-1}'' are the actual stress concentration factors at the same sections. With $N' = N''$, after simplifying,

$$\frac{A''}{K_{-1}''} = \frac{A'}{K_{-1}'} \text{ or } A'' = A'\,\frac{K_{-1}''}{K_{-1}'} \qquad (31)$$

For a cylindrical form of shank, diameter of the shank is

$$D_s = D_K\sqrt{\frac{K_{-1}''}{K_{-1}'}} \qquad (32)$$

where D_K is the minor diameter of the screw thread. Equations 31 and 32 are based upon the assumption that the line of the limiting nominal stresses in the Goodman diagram, *Fig.* 5, for the cross-section I—I, as well as the same line for the cross-section II—II are parallel to the line zero to σ_s. This is true for cross-section I—I, but may be an approximation for cross-section II—II. The values A'' and D_s from Equations 31 and 32 must be considered as the theoretical minimum shank cross-section area and diameter, respectively. For practical purposes the shank diameter should be somewhat greater than this minimum value to assure that the shank will not become weaker than the threaded portion of the bolt. The extent to which the shank cross-sectional area can be decreased is also limited because the magnitude of $\sigma_m'' + (\sigma_{-1}/K_{-1}'')$ must not exceed the magnitude of σ_s for the bolt material. Furthermore, shearing stresses in the shank due to torsional moment increase rapidly as the shank diameter decreases. Therefore, if the diameter of the shank is much smaller

than the thread diameter, normal and shearing stresses arising in the shank when the nut is tightened must be checked. For bolts which are not relieved from the torque, the safety factor N'' must be determined using Equation 28. This factor must be somewhat greater than the factor N' determined using the same formula.

Summary: The new calculation method suggested for bolted assemblies subjected to repeated loading considers actual forces acting on the bolts, their range ratio, and the influence of stress concentration and local stresses on the strength of the bolt. It is the author's opinion that use of this method can increase the accuracy and reliability for determining the strength of bolts. A number of cases taken from practical experience bear out this opinion.

Data available in literature concerning the influence of stress concentration and local stresses on the strength of the bolts make it possible to apply the method suggested. However, values of stress concentration factors given in TABLE 1 must be considered as preliminary. In order to improve and to extend knowledge in this field, further experiments must be conducted. The influence of such factors as the kind of thread, form of nut, material, production method, and size of bolts has to be investigated. Stiffness of connected parts in bolted assemblies also has not been sufficiently investigated, and must be studied both experimentally and theoretically.

The suggested method gives the basis for selecting the best relationship between thread diameter and shank diameter that will give the optimum design strength for a given basic size and weight of bolts and connected parts.

Influence of bending stresses on the fatigue strength of bolts, which hardly can be considered numerically in strength calculations, apparently affects strength of some bolted assemblies significantly. Dependence of bending stresses at the threaded bolt portion upon shank bending stiffness is shown, and the best cross-sectional form for the shank from the viewpoint of bending stress is also discussed.

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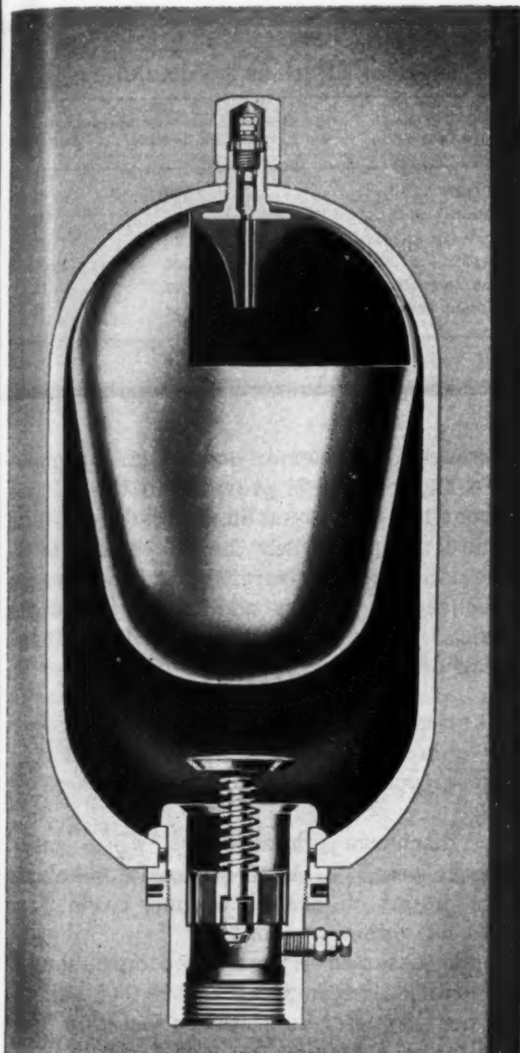
Beta Gage Speeds Inspection

AS A CONTINUOUS sheet of rubberized tire fabric leaves the calender rolls at the Armstrong Rubber Mfg. Co., it is automatically monitored for weight and thickness by a beta-ray gage. The sensitive device scans the strip of rubberized cord with beta rays emitted from a tiny source of strontium-90, a byproduct of atomic reactors at Oak Ridge, Tenn. Former procedure was to cut small square samples from the material as it emerged from the rolls. By the time these had been checked for weight and thickness, a large quantity of off-tolerance material had often been processed. The present procedure uses the amount of rays absorbed by the fabric as an indication of the material's weight per unit area. A deviation indicator on the General Electric instrument allows the calender operator to maintain the proper material gage.



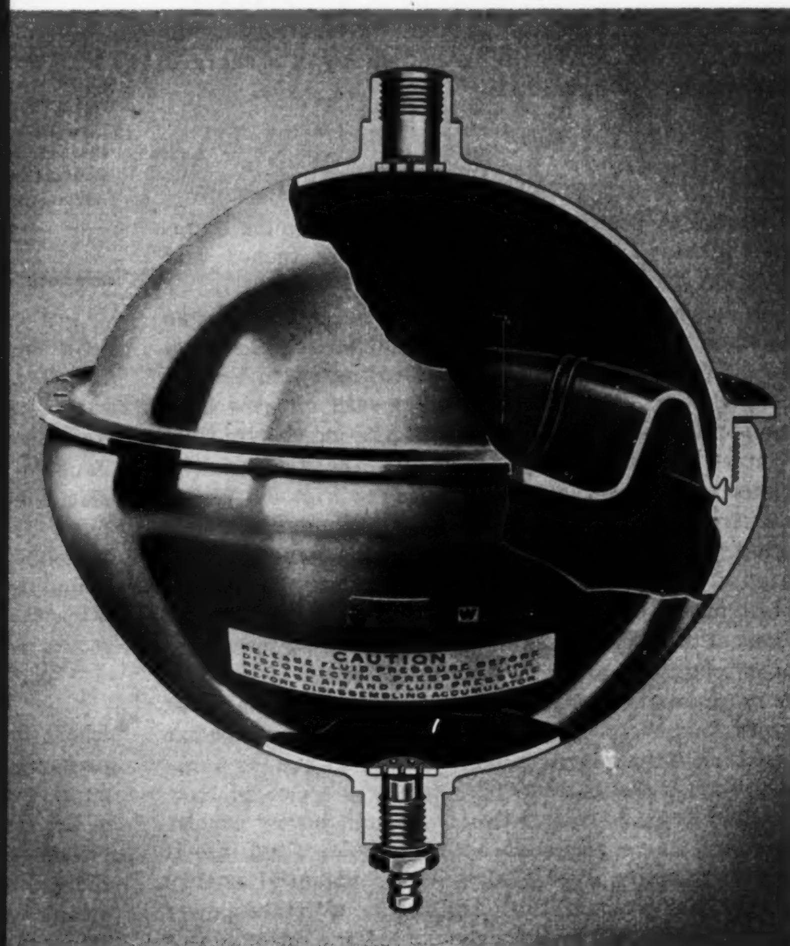


Fig. 1—Typical hydropneumatic accumulators. Bladder-type, left, made by Greer Hydraulics Inc. and diaphragm-type, below, made by Vickers Inc. may be precharged with either air or gas

Utilizing Accumulators in Hydraulic Systems

By Harold K. Palmer
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. . . how to determine accumulator capacity for optimum performance

MAXIMUM efficiency is achieved in many hydraulically operated machines through the application of accumulators at strategic locations in their hydraulic systems. Perhaps the most well-known accumulator function is that of storing hydraulic fluid under pressure to alleviate intermittent peak-demand pump requirements. Often, too, they are employed in a precharged state for emergency or standby service to insure safe operations in the event of partial circuit failure or pump disablement. Shock absorption, pulsation damping, static pressure-volume compensation, and pressure transfer between unlike fluids also are important functions of many hydraulic accumulators.

The original method of "accumulating" hydraulic power was to pump oil into a large cylinder contain-

ing a piston weighted with cast iron or concrete. This, of course, was a bulky and cumbersome arrangement, frequently subject to erratic sticking and leakage. However, the pressure potential of a "deadweight" accumulator is constant, making the analysis of its effect very simple. The potential energy of a charged deadweight accumulator is

$$W = \frac{d^2lp}{15.3} \qquad (1)$$

where W = potential energy, lb ft; d = piston diameter, inches; l = length of piston stroke, inches; and p = pressure of stored fluid, psi.

In modern accumulators, compressed air or gas is employed to pressurize the fluid vessel, the compressibility of the air or gas providing space for the fluid charge. Since air in contact with oil at a high temperature causes oxidation of the oil, air-to-fluid contact is avoided by separating the two with a piston, bladder or pliable diaphragm, *Fig.* 1. Where construction of the accumulator permits, nitrogen often is used instead of air. Pistons generally are used in the largest accumulators and pliable diaphragms or bladders serve as barriers in the smaller sizes. Piston-type accumulators usually suffer a loss of efficiency of about 15 per cent because of friction. A much smaller loss exists in diaphragm and bladder types, but their efficiency, nevertheless, is somewhat less than 100 per cent.

Operating Characteristics: Although power usually is stored in an accumulator by pumping fluid into it under pressure, it actually is the potential energy of the gas volume that fulfills the ultimate power demand. Therefore, the behavior of the accumulator must be studied from the standpoint of gas compression. If gas decompression is required during only a fraction of the cycle, the pump capacity insofar as recharging the accumulator is concerned need be only a corresponding percentage of the motor demand. However, the pump must exceed normal fluid motor operating pressure to store the maximum amount of oil required in the accumulator.

Table 1—Heat Effect in Accumulators

Conditions (100 cu in. accumulator)	Compression: Isothermal	Compression: Adiabatic
Initial pressure (psi)	500	500
Initial fluid volume (cu in.)	10	10
Final fluid volume (cu in.)	20	20
Final pressure (psi)	562	660
Pressure increase (per cent)	12	32
Average power input increase (per cent)	6	16

In the pressurized bladder or diaphragm accumulator, *Fig.* 1, a check valve is provided in the oil passage to prevent the bladder or diaphragm from being forced out into the pipe when the oil volume is exhausted or drained out. A service valve on the gas side for precharging permits adding gas to supplement the initial charge, or to make up for leakage. The operating or discharge pressure of such an accumulator is

$$p_d = p_i \frac{V - q_i}{V - q_r} \qquad (2)$$

where p_d = discharge pressure, psi; p_i = initial pressure charge, psi; V = total accumulator volume, cu in.; q_i = initial oil charge volume, cu in.; and q_r = reserve oil volume, cu in.

This formula assumes isothermal compression of the gas—a justified assumption because of the uncertainty of heat transfer and leakage losses in operation of hydraulic systems.

If adiabatic compression is assumed, the preceding formula becomes:

$$p_d = p_i \left(\frac{V - q_i}{V - q_r} \right)^{1.4} \qquad (3)$$

The exponent 1.4 is a mean constant for adiabatic compression of nitrogen and air alike. Obviously, if

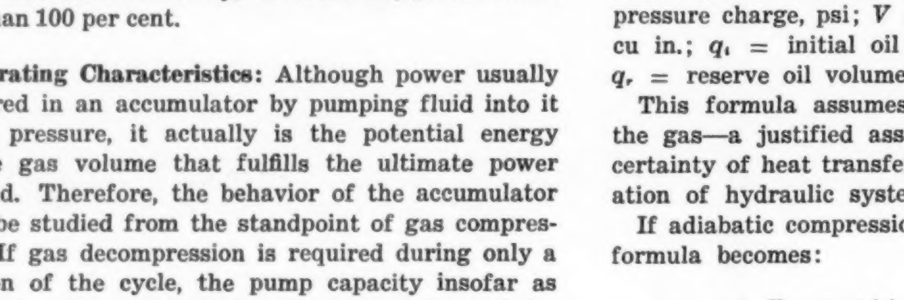


Fig. 2 — Simple mass curve representing accumulator operation of a single motor through its cycle. Shaded area represents fluid volume of the accumulator

Table 2—Cycle Phase Oil Demand*

Phase of Cycle	Working Load (lb)	Piston Diam (in.)	Piston Stroke (in.)	Phase Length (sec)	Hydraulic oil Pressure (psi)	Hydraulic oil Demand (cu in.)
A–B	5000	3	20	10	714	140
B–C	0	.	..	3	...	0
C–D	2000	2	5	2	637	15
D–E	0	.	..	3	...	0
E–F	1500	3	20	3	650	46
.	500	2	5	3	650	4
F–G	0	.	..	15	...	0

* From Example 1, using a 1.52 gpm pump.

Table 3—Cycle Pressure Fluctuations*

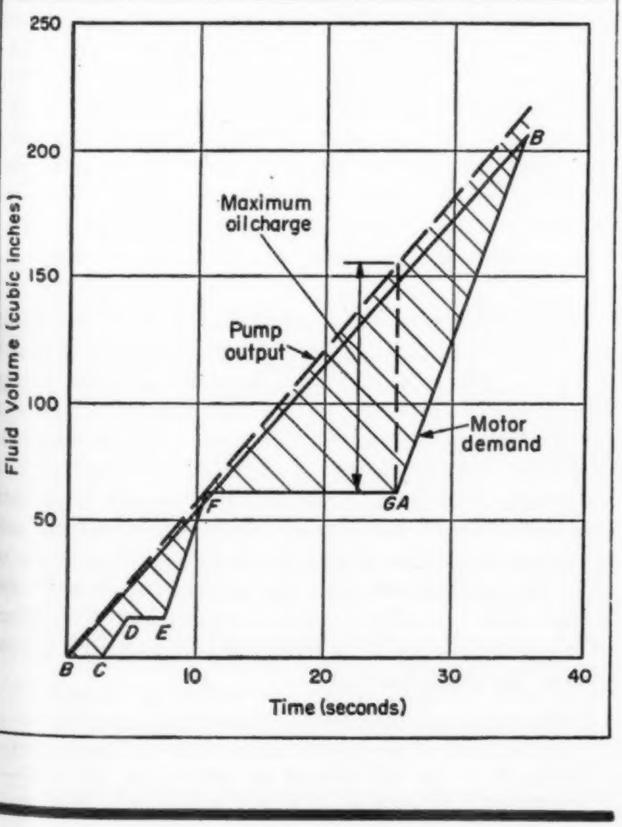
Point of Cycle	Oil Demand (cu in.)	Available Pressure (psi)	Required Pressure (psi)	Excess Pressure (psi)
A	90	875	714	161
B	5	714	714	0
C	25	738	637	101
D	20	730	637	93
E	32	750	650	100
F	5	714	650	64

* From Example 1, using a 1.52 gpm pump and a 2-gallon accumulator. Isothermal compression is assumed.

adiabatic compression is assumed, a greater accumulator pressure variation exists as compared with isothermal conditions. Typical difference in charged accumulator pressures based on the two methods of computation are given in TABLE 1. These values are for a 100-cu in. accumulator having an initial gas charge pressure of 500 psi. An initial oil volume of 10 cu in. is assumed to have been increased hydraulically to 20 cu in.

It will be noted that the fluid pressure after the oil volume increase is 12 per cent more than originally by the isothermal computation, and 32 per cent more by the adiabatic. Since the pressure increase is gradual during the oil pumping, the average power requirement increase while pumping will be 6 per cent for isothermal and 16 per cent for adiabatic.

Fig. 3—Mass curve representing accumulator operation of multiple-motor system through a complete cycle. Points B to GA and GA to B represent concluding and leading phases, respectively, of consecutive cycles



Graphical Analysis: For a specific hydraulic system the required pump capacity as well as the accumulator size may be determined simply by drawing the mass curve, as is done in water-storage problems. The mass curves will represent the integrated in-flow and out-flow from the accumulator. The difference between the two curves represents the amount of fluid in the accumulator. Gas content of the accumulator, the active element which neither flows into or out of the accumulator, changes its volume and pressure as the amount of fluid varies. A simple curve, *Fig.* 2 represents a single pump in conjunction with an accumulator serving a single motor.

In *Fig.* 2 the motor demand is integrated through two cycles so two maxima, A and C will cover one cycle. Line ABC represents the motor demand during one cycle, AB being the interval of no demand and BC the motor demand of one operation of the motor. Line AC represents the pump supply, the rate of which amounts to $C'C/AC$. The amount of fluid in the accumulator at any instant is the difference in the ordinates of AC and ABC at that instant. The maximum fluid content amounts to BB' at B when $BB' = (CC')\ (AB/AC')$.

When more than a single motor is to be handled by one accumulator a mass curve should be drawn and will prove sufficiently accurate to furnish essential data, considering the uncertainty in the pressure ratios. It also presents a clearer picture of the operation than is possible by tabulation alone.

EXAMPLE 1: Two double acting pistons are to operate consecutively in a cycle. First to operate is a 3-inch piston under a 5000-lb load through a 20-inch stroke in 10 seconds. Three seconds after this, a second piston 2 inches in diameter and under a 2000-lb load operates through a 5-inch stroke in 2 seconds. Following a 3-second interval, both pistons are reset simultaneously in three seconds. Reset load on the 3-inch piston is 1500 lb; on the 2-inch piston, 500 lb.

Data for this hypothetical cycle are set-up in TABLE 2. Hydraulic volume transfer during one complete cycle may be readily visualized from the corresponding mass curve, *Fig.* 3.

From TABLE 2, 205 cu in. of oil will be required

during the 36-second cycle. The average rate of demand, therefore, is 5.7 cu in. per second. Actually, the rate of oil delivery from the recharging pump should exceed the specific requirement to allow for leakage, etc. Assuming no fluid loss, in Example 1 a 1.52-gpm pump, with bypass control, would pump for 34 seconds and then run idle for 2 seconds.

Thermal Considerations: If the accumulator is called upon to supply power several times a minute, adiabatic expansion of the gas content probably should be taken into account; otherwise, isothermal conditions could be justifiably assumed. Furthermore, overall leakage from the system constitutes another uncertain factor to be considered.

The accumulator will be cold when first started after the machine has been shut down overnight and the first heat of compression will be dissipated rapidly through cold oil and the accumulator casing. Although a variation in pressure during warm-up periods is inevitable, excessive variation results in power waste and difficulty in controlling piston speed. At the same time, the minimum pressure should not fall below the minimum cycle requirement. Practical considerations, because of temperature fluctuations, make it advisable to limit the maximum oil content of an accumulator to about 125 to 150 per cent of the minimum, where little pressure variation can be tolerated. In this range the resulting adiabatic pressure would be 112 to 117 per cent of the isothermal pressure.

The accumulator itself must be relatively large to limit the pressure range. When size, weight, and efficiency are critical, it is advisable to design or select the accumulator with care; otherwise, if the total volume of the accumulator is about five times the maximum amount of oil to be stored, the size and efficiency will be suitable for average applications and a slight deviation from this size will make little difference in the final results. Some manufacturers recommend 6 to 11 times the volume of oil required as the nominal accumulator capacity.

Applying to Example 1 the rule that the volume of the accumulator be 5 times the peak oil demand of 90 cu in., from *Fig.* 3, the total volume amounts to 450 cu in. A commercial 2-gallon or two 1-gallon units would provide 462 cu in. and the relationship between oil volume and pressure for this accumulator is as expressed by Equation 2. *Fig.* 2 and TABLE 2 show that at point B the pressure should not fall below 714 psi and the reserve oil volume, q_r, should not be less than 5 cu in. to allow for leakage.

The pressures at various points of the cycle as determined through Equation 2 are shown in TABLE 3. The last column headed "excess pressure" shows the difference between accumulator and required pressure. This denotes power inefficiency resulting from use of an accumulator for several functions at different pressure levels. Only in cases where the accumulator leads to added convenience or a saving in weight or space can its use be justified for such installations.

Power Consideration: In planning hydraulic systems, designers should consider the effect of accumulators on the power input requirement. A convenient expression of power input is:

$$P_i = \frac{Qp}{1714.4\,E} \qquad (4)$$

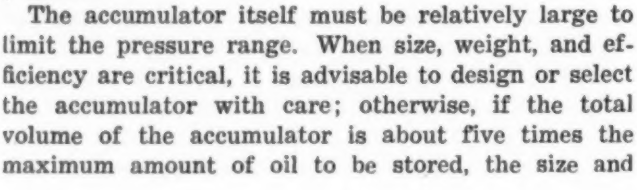
where P = pump power input, horsepower; Q = pump discharge, gpm; p = discharge pressure psi; and E = pump efficiency, per cent.

In the previously discussed accumulator circuit the pump capacity established was 6 cu in. per second. or 1.52 gpm. Maximum pressure against which the pump operated in charging the accumulator was 875 psi. Assuming isothermal gas compression in the accumulator and a pump efficiency of 90 per cent. the maximum power input is 0.88-hp. In contrast operation of the same system without the aid of an accumulator would require a minimum 3.64-gpm pump with a continuous power requirement of 1.68 hp.

Shock Absorption: An accumulator is a convenient accessory as a shock absorber. Water-hammer shock is very much relieved if the deceleration time of the moving fluid column can be lengthened by only a few milliseconds. For this purpose the accumulator is highly beneficial for it absorbs the flow with a gradually increasing pressure and avoids a sudden rise. The single shock-wave cycle being very short. a matter of a few milliseconds only, the volume of fluid to be injected into the accumulator is slight. which calls for only a small accumulator. The effect of an accumulator in reducing shock pressure is best shown by the following:

EXAMPLE 2: A circuit under 1000 psi pressure contains 10 feet of pipe, 0.312-inch inside diameter, carrying 20 cu in. per second or 0.02 cu in. per millisecond at a velocity of 21.6 feet per second. With no

Fig. 4—Shock-pressure curve developed from Example 2. Under conditions outlined, the curve indicates peak-pressure relationships with fluid deceleration time



accumulator the shock pressure due to sudden stoppage is $53v = 1150$ psi, in addition to the normal pressure of 1000 psi, or 2150 psi in all. Or,

$$p' = 53\,v + p = 2150 \qquad (5)$$

where p' and p are peak pressure and normal pressure respectively, in psi; and v = oil velocity, feet per second. The factor 53 is a constant for oil with a specific gravity of 0.8 and a modulus of compressibility of 250,000.

When a moving column of oil is stopped by the closure of a valve downstream, the line-pressure surge* in psi is, empirically,

$$p = \frac{49\,v}{4500\,\dfrac{T}{L} - 1} \qquad (6)$$

where T = valve-closing or oil-deceleration time, seconds; and L = length of oil column in feet.

Applying the assumed circuit data of Example 2, Equation 6 becomes

$$p = \frac{(49)\ (21.6)}{4500\,\dfrac{T}{10} - 1} \qquad (7)$$

Total pressure, p_t, within the line, of course, then is

$$p_t = 1000 + \frac{1058}{450\,T - 1} \qquad (8)$$

from which the pressure curve in *Fig.* 4 is plotted.

If it is desired to limit the peak pressure in 0.0125 second stopping time to 1225 psi by means of an accumulator. Then, assuming the initial oil volume $q = 0$, total volume of the accumulator, from Equation 3, is expressed in

$$p = 1225 = 1000\left(\frac{V}{V - q}\right)^{1.4} \qquad (9)$$

Then

$$\frac{V}{V - q} = 1.16$$

from which $V = 7.25\,q$.

Next, conditions during stoppage of the oil can be described by the equation

$$q_2 - q_1 = q_2 = 0.0125\left(\frac{20 - 0}{2}\right) = 0.125 \qquad (10)$$

That is, the volume of oil entering the accumulator is simply the product of the time period and the mean velocity effective during that period. Or,

$$V = 7.25\,q_2 = (7.25)\ (0.125) = 0.91 \qquad (11)$$

Therefore, to reduce the shock pressure from 2150 to 1225 psi requires an accumulator capacity of 0.91 cubic inches.

*"Water Hammer in Hydraulic Systems"—Harold K. Palmer, MACHINE DESIGN, September, 1952, page 175.

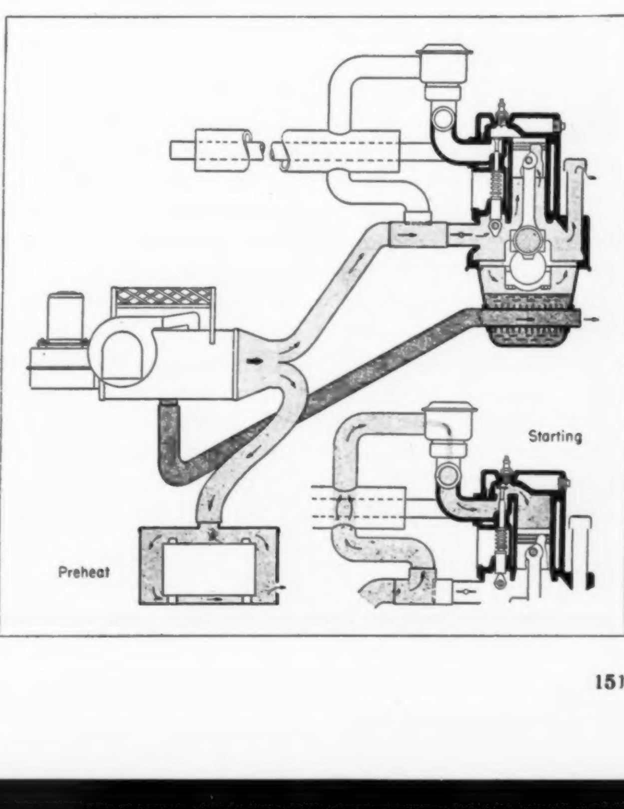
Engine Preheater

ENGINE preheating equipment, which makes starting of diesel or gasoline engines possible under severe arctic conditions in what is claimed to be the shortest time yet attained, is now in production at Stewart-Warner Corp. Design of such heaters, requiring quick engine starting at −65 F, has three basic objectives: (1) Resistance to cranking power required must be minimized to permit the engine starter to turn the engine fast enough for reliable starting in minimum time; (2) some method must be provided to augment the normal engine ignition system during starting periods, since engine fuels are difficult to ignite at −65 F; (3) lubricating oil must be heated sufficiently to promote the circulation and to provide normal lubrication when the engine is started.

The interior of the engine is heated, and congealed oil on the friction parts of the engine is warmed, by circulation of clean air heated to approximately 400 F through the engine. Tests on systems of this type have resulted in reduction of resistance to cranking power sufficient to enable easy cranking in from 10 to 20 minutes' heating time.

Lubricating oil may be heated by means of a heat exchanger built into or around the oil sump. A successful method used in tests involves finned tubing routed through the oil sump, with 400-degree air.

Using a standard vehicle heater with a normal delivery of 20,000 Btu per hour, a General Motors 6-cylinder, 2-cycle diesel engine weighing over 3000 lb has been started with less than one minute's cranking at 60 to 90 rpm at temperatures from −25 to −65 F, using SAE 10-W oil. This engine was preheated from 25 to 75 minutes. Cranking power inputs ranged from 6000 to 10,000 watts.



DESIGNING WITH PLUG WELDS

Fig. 1 — Drillpress table designed for plug welding reduced weight 8 pounds and cost 25 per cent over previous design

By Omer Blodgett
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EXCELLENT design results can often be obtained with "plug" welds, an area of welding often overlooked. Such welds are made by filling holes in one part with metal welded-in. With proper forethought in design, most of these welds can be hidden so that the grinding of welds for external appearance can be eliminated to distinct cost advantage.

An excellent example of such design is the Snow Mfg. Co. drill table, *Fig.* 1, and the vertical way supports for both the table and drillhead. The pan or base portion of the table is drawn from $\frac{3}{16}$-inch plate. Two $\frac{3}{8}$-inch steel plates, spaced $\frac{3}{8}$-inch apart, form the work surface of the table. Each plate is supported and stiffened with $\frac{3}{16}$ by $1\frac{1}{2}$ by 4-inch channel which is plug welded both to the work plate and to the pan, *Fig.* 2. The slot created by this assembly accommodates T-bolts and no machining is necessary owing to absence of fillet welds along the corner joints.

The dovetail plate, *Fig.* 2, is not made from a single piece but is built-up from a bevel-edge bar and a bar of square section, both plug welded to the back plate. These components are easily and quickly finished to requirements by grinding all critical fit surfaces after welding.

Slideway pieces for the table and drillhead are fastened to the front of the formed steel column by the same method. Plug welds are made in holes predrilled in the column face, *Fig.* 3. Cold-rolled strips of $\frac{1}{4}$ by $1\frac{1}{8}$-inch section are clamped over the rows

of holes and the latter are filled with weld metal applied easily from the back. These welds fasten the strips securely and do not show on the exposed surfaces, which are ground all over after welding.

The invisibility, low cost and ease of production afforded by plug welds should be seriously considered wherever strength requirements permit. Specifications for both plug and slot welds for correct application are available* in the literature.

*1. *Production Processes—Their Influence on Design*, Volume Two, R. W. Bolz, Penton Publishing Co., Cleveland, 1951.
2. *Welding Handbook*, American Welding Society, New York, 1951.

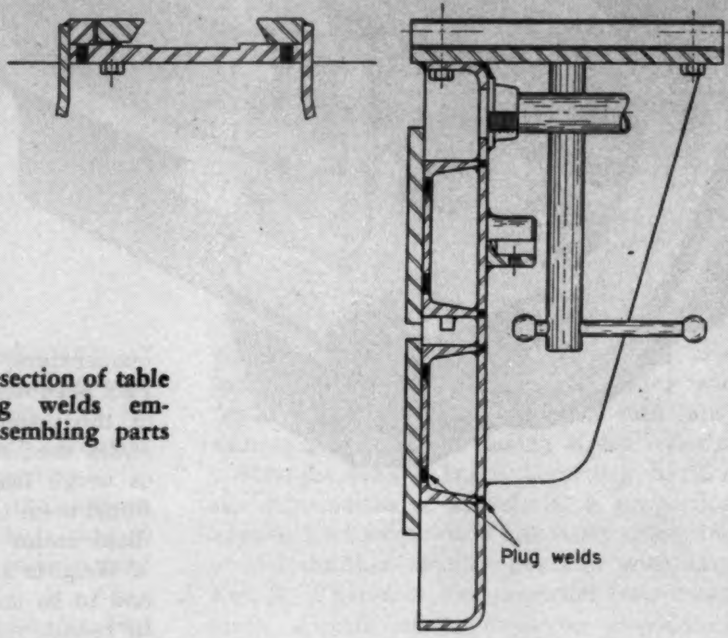


Fig. 2—Cross section of table showing plug welds employed for assembling parts

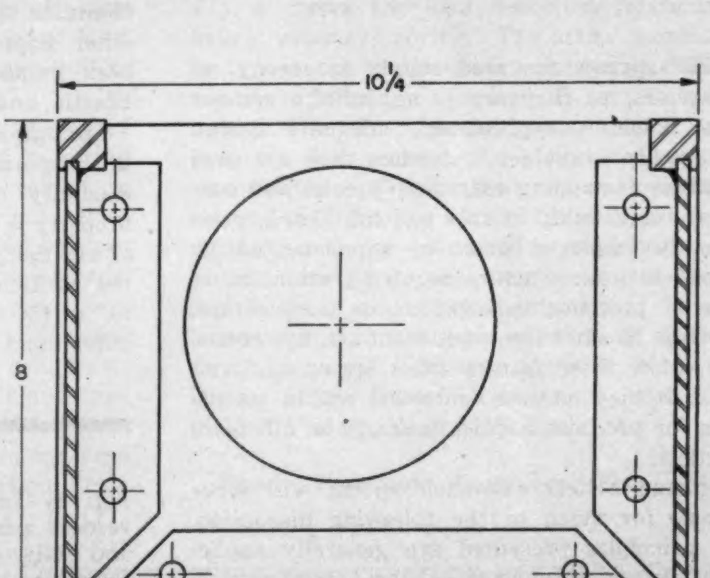
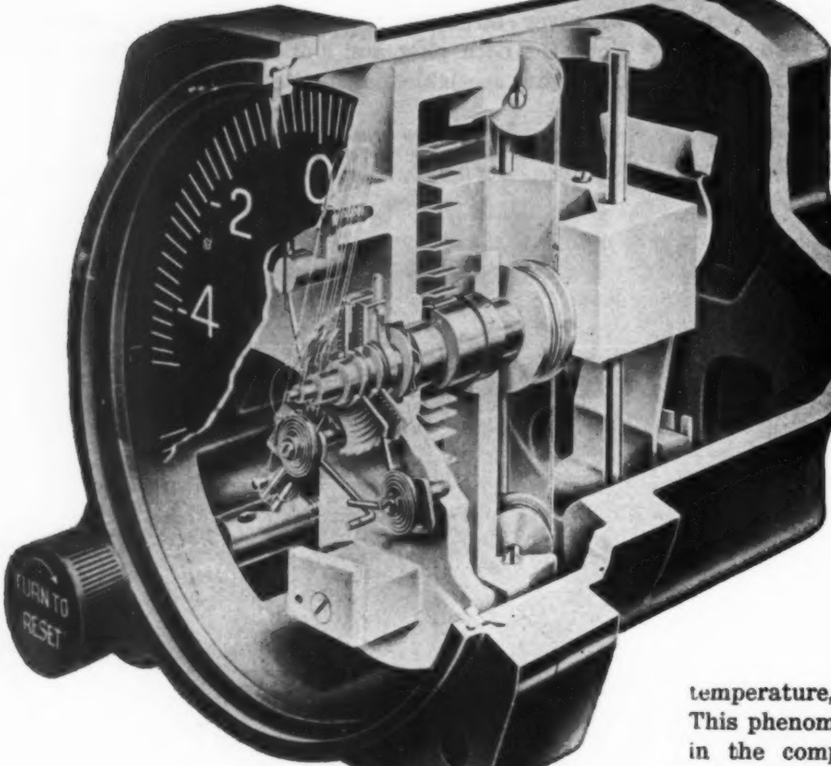


Fig. 3 — Sectional view through column of machine showing attachment of slideway pieces with plug welds



By M. Gerard Fangemann
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New York .

When springs are used simply as energy reservoirs, as they are in an infinite number of machine applications, adequate design rules are readily available. But when they are used in applications demanding extremely precise and constant linear relationship of rate to load, *Fig.* 1, principles usually employed must be supplemented by other considerations. Factors requiring attention in the design of precision springs include temperature, deviation from Hooke's law, permanent set, hysteresis, and creep. How these factors affect spring performance and how they may be controlled within acceptable limits for precision applications will be discussed in this article.

The common helical extension spring will serve as the basis for much of the following discussion. However, principles presented are generally applicable to all types of springs — helical compression, spiral, leaf, etc. *Fig.* 2, for example, shows a compression spring application.

Temperature Compensation: Springs of special alloys that inherently maintain essentially constant stiffness throughout a temperature variation range are known as temperature-compensated springs. Steel springs vary in stiffness when subjected to changes in temperature; they become weaker with increasing temperature, stiffer with decreasing temperature. This phenomenon can be serious, as illustrated clearly in the computing scale industry. If food-weighing scales used steel springs, a true pound would appear to weigh less in a cold atmosphere and conversely would seem to weigh more at high temperature. Since these scales are subject to approval of the Bureau of Weights and Measures, costly thermostatic devices had to be used in the past to correct this condition. In recent years temperature-compensated alloys, essentially of the Elinvar type, have been developed to eliminate the thermostat type compensators, and other improved characteristics have simultaneously been gained. Two materials of this type are Iso-Elastic and Ni-Span-C, Table 1.

Springs remain uniform in stiffness with variation in temperature primarily because the modulus of elasticity of the material remains constant. This property is the temperature coefficient of the modulus of elasticity. For the Iso-Elastic alloy, for example, the coefficient varies from -20×10^{-6} per deg F to $+ 15 \times 10^{-6}$ per deg F. Spring steel, on the other hand, has a temperature coefficient of $-190 \times$

Fig. 1—Top—Acceleration of an aircraft along its vertical axis is indicated through the precision spring and pulley system of this Eclipse-Pioneer accelerometer. Operation over a temperature range of -55 C to 70 C and minimum hysteresis and creep are demanded of the spring material

Fig. 2—Right—Magnesyn unit, shown exploded, transmits aircraft oil or fuel pressure data electrically to flight engineer's station. Iso-Elastic compression spring shown is corrosion resistant and maintains load-deflection ratio within 1 per cent from -67 F to 158 F

PRECISION SPRINGS

What factors must be considered in their design and application?

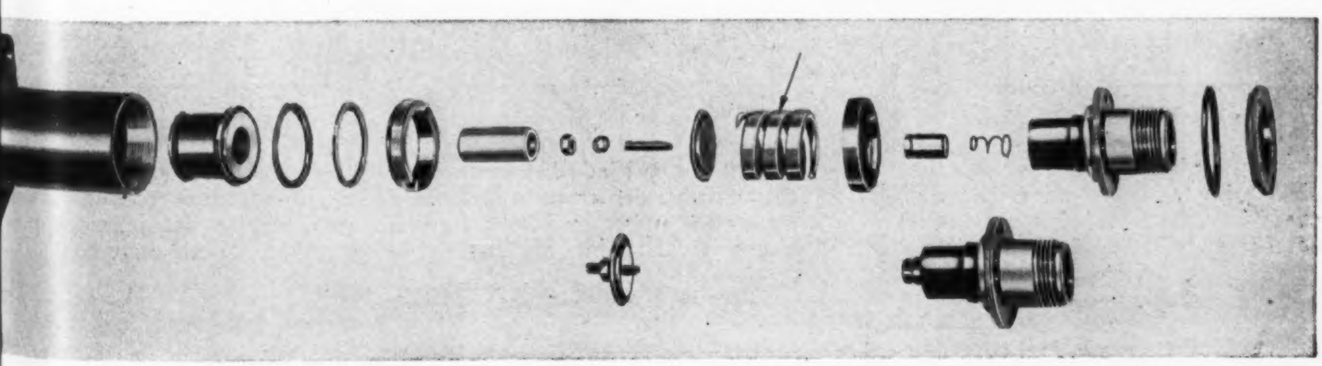
10^{-6} per deg F. The sign of the thermal coefficient indicates the direction of change in strength. A plus sign means that the spring rate increases with increase in temperature, and a minus sign indicates a weakening of the spring rate with temperature increase. Spring rate, of course, is simply the ratio of load to deflection; that is, a spring having a rate of 10 pounds per inch will deflect 1 inch when a load of 10 pounds is applied.

The recommended range of temperature for Iso-Elastic is from -50 F to $+150$ F. However, if drift and hysteresis tolerances are enlarged slightly, this range can be increased. The temperature coefficient of the modulus of elasticity will not change noticeably until 300 F is exceeded.

Typifying one type of temperature-compensated spring material, Iso-Elastic receives most of its spring properties from cold reduction in area; it cannot be hardened by heat treatment. Area reduction varies according to the finished size, but is usually between 87 and 93½ per cent. After it is formed into springs, the material is given a special low-temperature stress relief (½ hour at 750 F). On the other hand, Ni-Span-C is a hardenable type alloy which requires heat treatment varying in temperature (1100-1350 F) and time (2-8 hours) according to characteristics desired. An important difference between the two alloys concerns the production of springs having initial tension. Since Ni-Span-C springs require hardening, most of the initial tension imparted during coiling will be lost. Iso-Elastic springs can be coiled with initial tension that is not sacrificed during stress relieving.

Straight Line Error: According to Hooke's law, the deformation of a material is proportional to the applied load when loads are fairly small, but the rate of deformation usually deviates with larger loads, *Fig.* 3. Therefore, for precision force-measuring devices, special means must be provided to assure straight-line relationships between load and deflection. *Fig.* 4 shows the load deflection relationship of a helical extension spring. The actual measured deflection does not start at zero normally, since there is usually a preload or tare load, P_0, on the spring. The straight-line error for any given load, P, is the discrepancy MN between the actual deflection and the straight line. Metallurgical and mathematical analysis showed that the straight-line error of a helical spring could be improved considerably with the proper selection of the cross-section of the material in relation to other spring dimensions. In particular, it was found that a ratio of width to thickness of 3 to 1, and a ratio of approximately 20 to 1 for the spring index (mean diameter to wire thickness) yielded a minimum straight-line error.

Plastic Flow and Permanent Set: Although a



spring may depart from Hooke's law slightly, the spring material in itself may be perfectly elastic. That is, no permanent distortion remains after complete removal of the load, providing the spring is not stressed beyond its safe limit.

There is, however, an elastic limit beyond which the material begins to distort permanently, so that with subsequent removal of the load only partial recovery results. This partial recovery is due to plastic flow under high stress and is called permanent set. The material still springs back but fails to return to its initial position, *Fig.* 5. Deviations in straight-line performance can therefore exist normally within the elastic limit of the material, or they may stem from a condition of overstressing that will lead to permanent set.

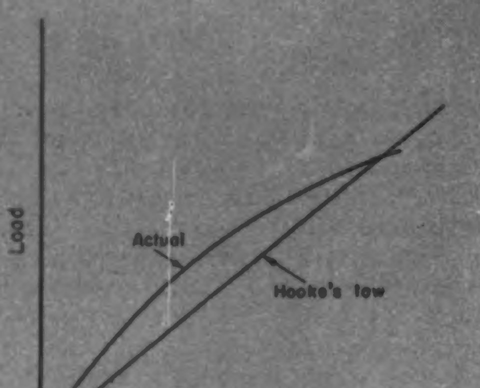


Fig. 3—Above—Load-deflection curve of many spring materials deviates from Hooke's law

Hysteresis: Mechanical hysteresis is well known in the scale industry as "back error." This phenomenon occurs during the removal of a previously applied load and is the lag of the deformation of an elastic material. The magnitude of deformation depends upon the material. For high-grade spring alloys, the hysteresis error ordinarily would be about 0.02 per cent of the total deflection. For some spring materials, the back error is usually 0.1 to 0.2 per cent of the total deflection. *Fig.* 6 illustrates the hysteresis loop resulting with a complete loading and unloading cycle of a spring. The deflection difference *MN* is the actual hysteresis error.

Explanation of hysteresis requires examination of the stresses and strains within the metal, since there is a small permanent deformation of the individual crystal grains. The deformation is elastic up to the elastic limit, whereupon the material begins to flow plastically. The crystal springs back elastically upon load removal; however, if a plastic flow existed, a permanent deformation would remain after removal of the load.

If a reverse load were applied to this deformed grain, the deformation would again be elastic but in the opposite direction to the initial deformation. In reality this would be a continuation of the elastic recovery during unloading. Actually a material is composed of many grains, and the measurable deformation is the summation of the individual deformations of each grain. Part of the grains would deform only elastically and would recover completely. However, other grains would be overstressed, flowing plastically, and would not recover. The variously stressed grains contribute in varying degrees to the observed deformation.

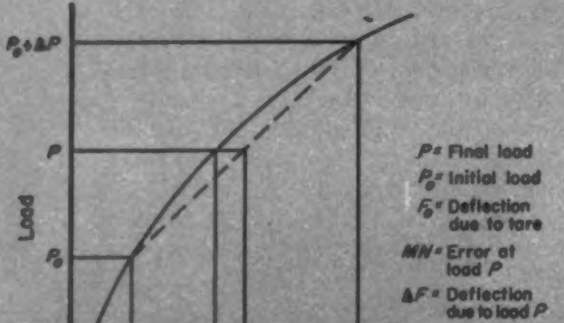


Fig. 4—Straight-line error of a calibrated spring at any load is the deflection increment *MN* between the actual spring curve and the straight-line joining the terminal points of the spring range

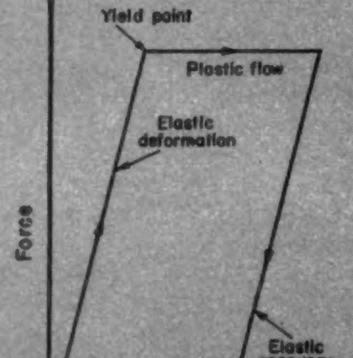


Fig. 5—Simplified diagrammatically, elastic deformation and recovery are offset by plastic flow above yield point of material

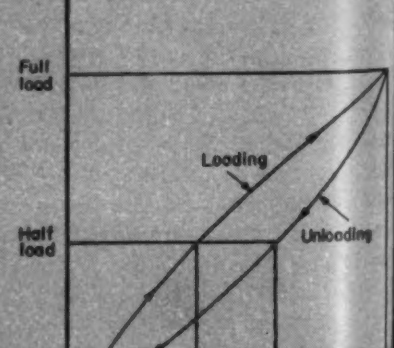


Fig. 6—"Back-error," *MN*, stems from hysteresis of the material, the lag in deformation of an elastic material

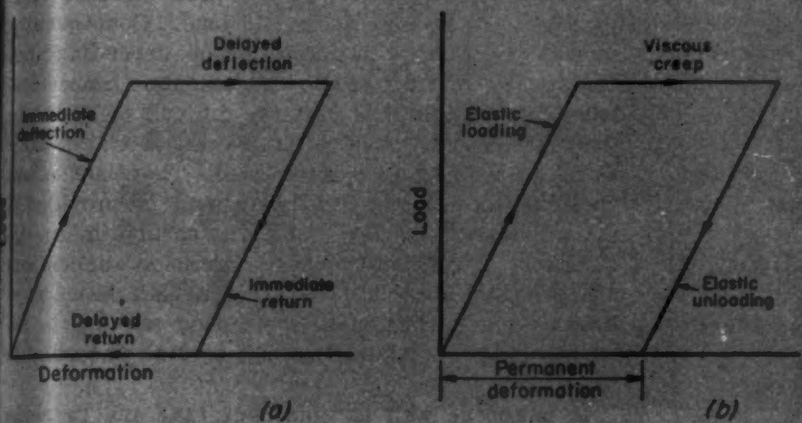


Fig. 7—Drift after load application, *a*, is recovered after load removal for some materials and conditions, but viscous creep, *b*, results in permanent set

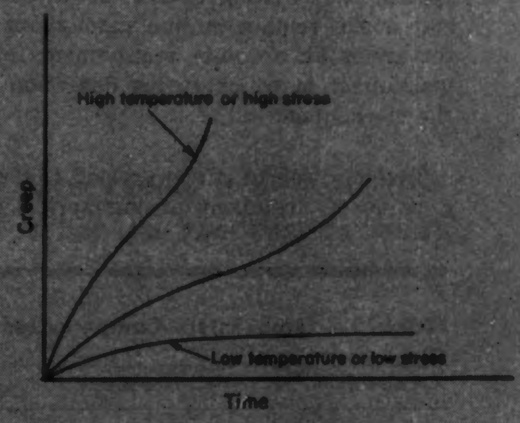
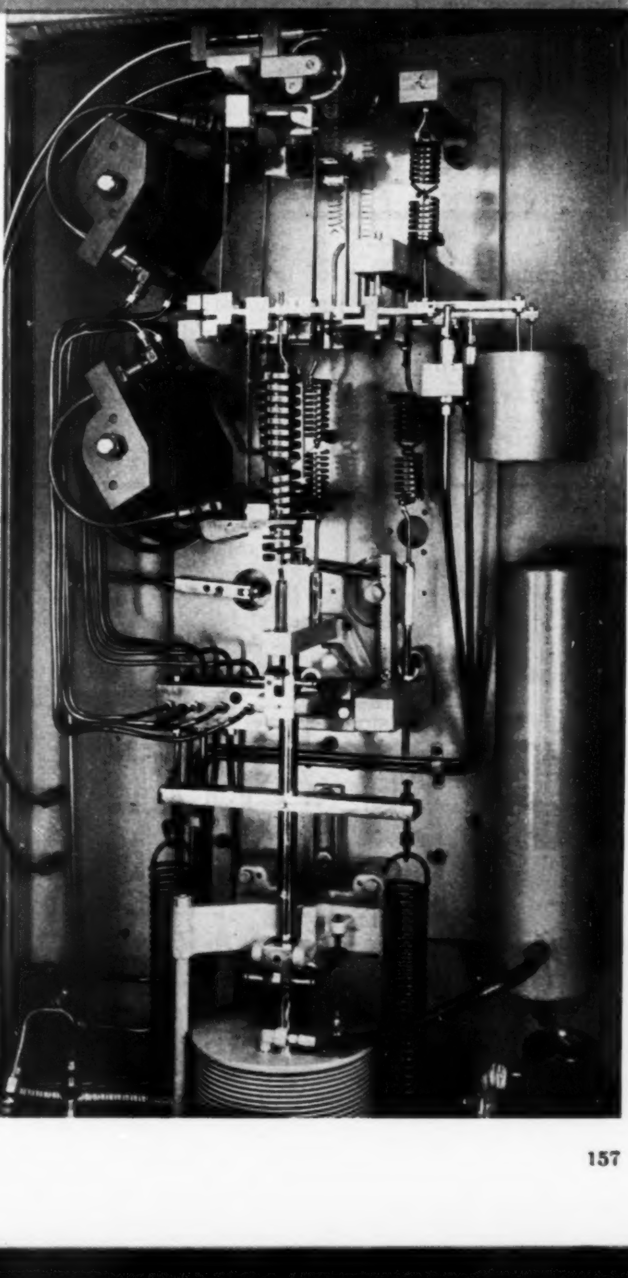


Fig. 8—Above—Drift or creep rates increase with higher temperature or higher stress

Fig. 9 — Below — Twin extension precision spring systems are used in this Tate-Emery weighing head for Baldwin-Lima-Hamilton testing machines. With spring twist neutralized by the reversed direction of coiling, straight-line error is less than 0.01 per cent, and combined creep and hysteresis errors are less than 0.3 per cent at midload



Drift: An important property of precision-spring materials is drift or creep. This phenomenon occurs when a spring is loaded and reaches capacity instantaneously. Although the spring appears to deflect instantaneously, actually a slight deflection occurs after load is applied. This minute deflection is only a small fraction of the total deflection of the spring and is dependent on the time the load remains constant. The rate of this elastic deformation is also dependent upon the fiber stress and work-hardening of the material that may take place. This deformation, termed drift, should not be confused with deformation producing permanent set when the elastic limit is exceeded. This phenomenon is illustrated in *Fig.* 7. Most of the deformation takes place in a relatively short period of time. With some materials, such as the Iso-Elastic alloy, the deformation occurs usually within a few minutes; however, with other materials, the deformation may continue for hours or even days.

When this phenomenon occurs at elevated temperatures, the deformation is more commonly termed creep. The greater the work stress and the higher the temperature, the more nearly viscous will be the character of the drift. When lower temperatures occur with low stresses, most materials behave elastically. *Fig.* 8 illustrates the influences of stress and temperature on the magnitude of drift rate.

Twin-Spring Systems: Another error occurs whenever a spring is deflected. In an ordinary coiled extension spring, the outside diameter of the spring necessarily decreases whenever extension occurs. Since the length of wire in the spring is fixed, there is necessarily a rotation of the end hooks during extension. The smaller the deflection, or the greater the number of coils, the smaller the rotation. Although an error so caused may be slight, it may be noticed in precision applications. At Chatillon, precision springs have been designed in two parts, one half coiled left hand and one half coiled right hand to

form equivalent springs, *Figs.* 9 and 10. In such a system the twisting actions are neutralized, and the end hooks remain in line throughout loading. This feature is an absolute requirement for any precision instrument in which spring deflection is magnified in the mechanism.

Tare or Preload: The preload of a spring is always a point of argument among spring and scale engineers, since the question of initial tension invariably arises because of space limitations. The greater the tare load, the greater the tare extension of a spring, *Fig.* 10. If springs were coiled with spaces between coils in the free state, the extension due to tare load would be quite large. Tare load extension plus the deflection due to load would require extreme axial space for the springs under full load. Consequently, springs are coiled with initial tension, depending upon the amount of tare in the instruments. Since close tolerances on initial tension are difficult to maintain, there necessarily must be a tolerance on the hook length of the spring with tare load. Therefore, allowance for this tolerance on length must be provided in the adjustment hooks holding the springs in the instrument. Usually, springs for precision applications are supplied by the manufacturer with adjustment hooks already coupled to the spring and set to the specified span distance within several thousandths of an inch.

Table 1—Temperature-Compensated Spring Alloys

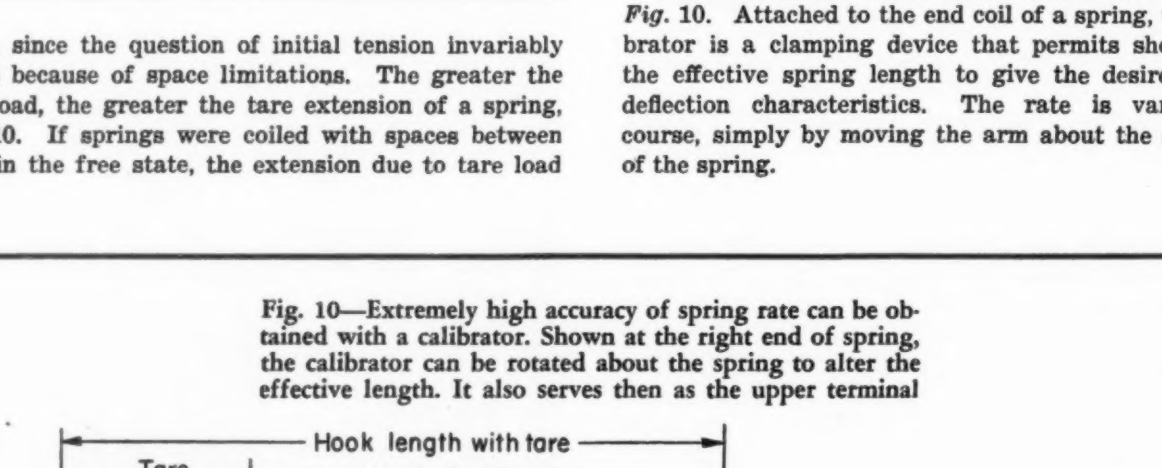
Property	Iso-Elastic	Ni-Span-C
Composition (per cent)..	36 Ni, 8 Cr, 0.5 Mo, bal. Fe	42 Ni, 5 Cr, 2.5 Ti, 0.06 C, bal. Fe
Tensile Strength (psi)...	170,000	90,000-200,000
Modulus of Elasticity (psi)	26,000,000	24,000,000-27,500,000
Torsional Modulus (psi).	9,200,000	10,000,000
Working Stress in Bending (psi)	90,000-100,000	15,000-110,000
Working Stress in Torsion (psi)	40,000-60,000	....
Thermal Coefficient of the Elastic Modulus (psi per deg F)	−0.00002 to + 0.000015	−0.00001 to + 0.00001
Hysteresis Error (per cent of deflection)....	0.05 max	0.05 max
Creep Error (per cent of deflection in 5 minutes)	0.02 max	0.02 max
Coefficient of Linear Expansion (in. per in. per deg F)	0.000004 approx.	0.0000045
Hardness (Rockwell) ...	C30-C36	B70-C44
Electrical Resistance (ohms per mil ft at 20 C)	528 approx.	480-580

It might be well to point out that any precision spring should be extended at least once to full extension in the instrument or scale application before the accuracy of the complete unit is checked. After the spring is subjected to its maximum extension, all adjustment hooks and parts will then be completely aligned, and the small amount of set that may have taken place in the adjustment hooks will also be eliminated.

Calibration: Every spring for precision weighing is calibrated to a specified rate. Accuracy of calibration depends entirely upon the application, since the cost increases considerably with the precision required. Springs can be calibrated to within several thousandths of maximum deflection if the need arises. Ordinary requirements for scale springs usually call for accuracy of 0.1 to 0.2 per cent in rate.

For exacting rate requirements, a calibrator is used. *Fig.* 10. Attached to the end coil of a spring, the calibrator is a clamping device that permits shortening the effective spring length to give the desired load-deflection characteristics. The rate is varied, of course, simply by moving the arm about the end coil of the spring.

Fig. 10—Extremely high accuracy of spring rate can be obtained with a calibrator. Shown at the right end of spring, the calibrator can be rotated about the spring to alter the effective length. It also serves then as the upper terminal



Impact Strength of Plastics

How should it be measured? What do the results mean?

By William Schack
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PLASTICS show quite different deformation characteristics than metals. It has been understood for a good many years that the Izod test, taken over from the impact testing of metals, was inadequate for measuring the same property in plastics.

Empirically, it was plain that something was amiss when the Izod value for mica-filled phenolic was markedly higher than for woodflour-filled phenolic, since in actual service the latter is slightly superior in impact strength. Further, as the resilience of a plastic increases, the less reliable does the Izod value become for comparative purposes. Thus, the General Electric Co. found it necessary to employ a magnetic drop-hammer tester to demonstrate the true superiority of their Hycar-phenolic blends over the straight phenolic compounds. By the Izod test, the woodflour-filled phenolic rated 0.3 ft-lb per inch notch; the woodflour-filled rubber-phenolic blend, 0.6. By the drop-hammer test, this 100 per cent differential was almost doubled for cracking the test specimen (0.924 vs. 3.330 ft-lb) and more than tripled for breaking it (1.508 vs. 9.480). Similar differences exist for the two phenolic types with other fillers.

In practical design, the drop-ball test has a double advantage: it is a truer index of impact strength than the Izod test, and it suggests an additional safety factor in the spread between the energy to crack and the energy to break, in such components as may still function in a cracked condition.

Theoretically, it has been demonstrated that the Izod value includes the energy to break the test piece, the energy to deform it plastically, the energy to throw off the broken piece, and finally the energy lost in the vibration of the machine. Since the only interest is in the first factor, which is the impact strength proper, any variables in the other three which affect the comparative values of the "total" impact strength as given by the Izod test make them misleading in effect. Several investigators have noted that the third factor—the energy to throw off the broken pieces (hence, called the "toss factor")—accounts for a large part of the nominal Izod value. If this could be eliminated, the resulting value would be closer to the real strength of the plastic.

That is the object achieved in a new testing machine, *Fig.* 1, developed by Rahm and Maxwell at the Princeton University plastics laboratory.[1] The test specimen is clamped to the edge of a bar-like rotor; its projecting edge is broken against a stationary anvil as the rotor moves freely after being brought up to the desired speed. The impact strength, or energy to break, is computed from the slowing up of the rotor as the break occurs. Since it can be proved mathematically that energy in the broken test specimen equals the energy in it prior to the impact, no energy is lost in the broken piece; i. e., the toss factor has been avoided.

The result is that the values obtained by this machine are lower than those derived from the Izod test, as is graphically shown in *Fig.* 2. Note that in some cases the new values are no more than half of the Izod figures. The toss factor varies markedly among the various plastics because of the differences in specific gravity: the denser the plastic, the more energy it absorbs as it is broken in the Izod apparatus.

Following up these investigations, Maxwell and

[1] References are tabulated at end of article.

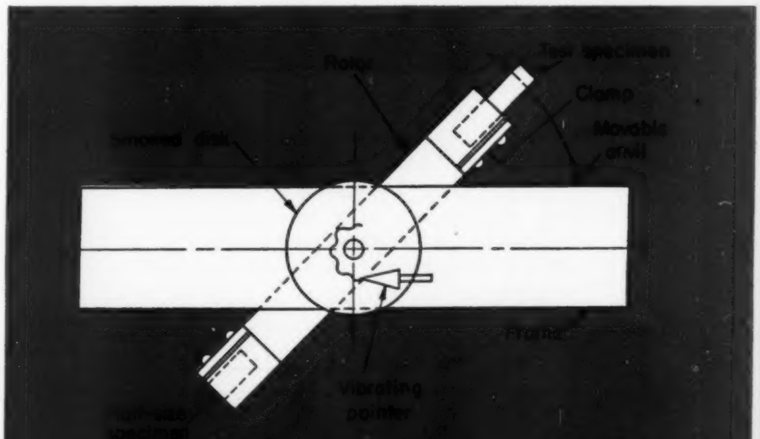


Fig. 1—Schematic of Rahm-Maxwell impact testing apparatus. Vibrating pointer measures angular velocity

Fig. 2—Comparison of Izod impact strength with results obtained by Rahm-Maxwell machine

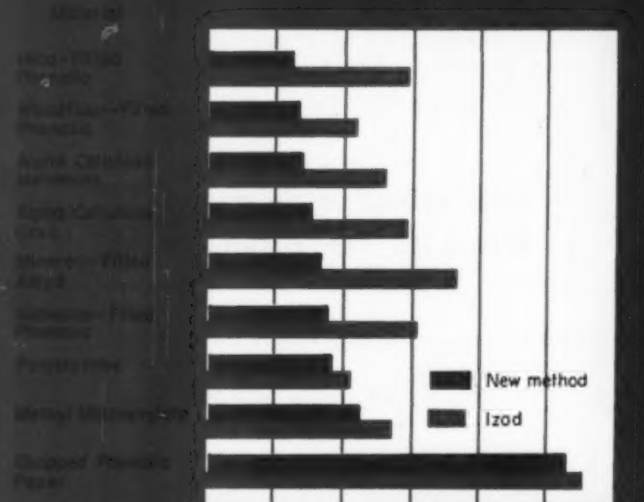


Fig. 3—Below—Load versus strain rate for polymethyl methacrylate. At each temperature, the maximum load increases linearly with the logarithmic rate of strain

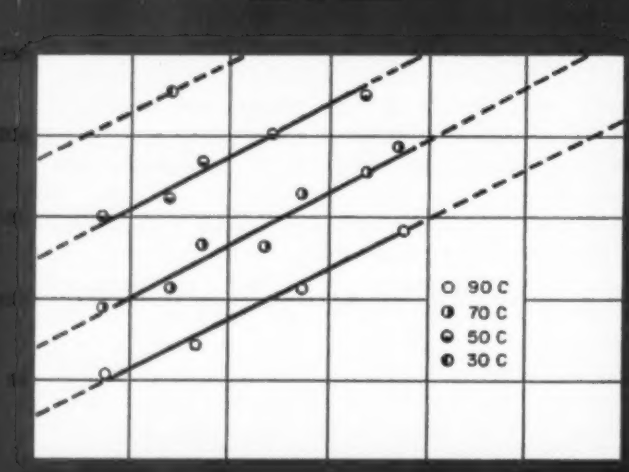
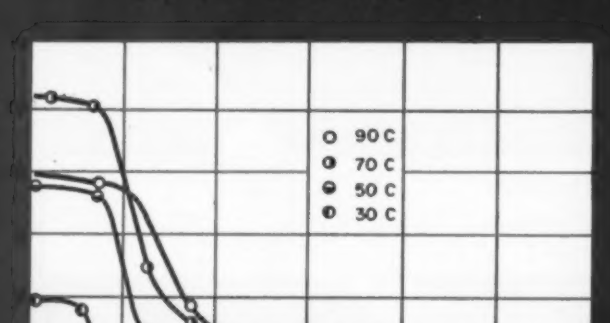


Fig. 4—Below—Energy to break versus strain rate of polymethyl methacrylate. Two distinct energy drops occur as strain rate increases



others at Princeton adapted the machine to more varied test conditions.[2] What happens as the rate of loading changes and as the temperature changes? This is the same problem which the designer must face when his product or component will not be in use only at room temperature and will not always be subjected to a single shock, and in the same direction. The original apparatus was modified because when a plastic component receives a sudden shock, the failure usually occurs in tension. That is, failure starts in that part of the component which is subjected to the highest tensile stress by the suddenly applied force. While properly designed sections and fillets can help to prevent breakage, the direction of the blow cannot be anticipated. It is therefore impossible to design the piece so that some section will not be put in tension if a shock is applied to the unit.

For this reason, it was decided that if the test specimen were subjected to a simple tensile stress rather than flexure, the results would be more directly applicable to practical problems as well as lend themselves more readily to theoretical analysis. To obtain such results, somewhat similar apparatus was devised to measure the energy to break at velocities of from 20 inches per sec to 300 or 40 inches per sec over a temperature range from 10 C to approximately 110 C. For tests at a velocity ranging from 0.0045 to 17.9 inches per second, a direct spring-operated load-weighing device was coupled with a mechanical elongation measuring system to give the complete load-elongation curve, and for the very low rates of straining a universal Baldwin testing machine was used.

The experiments carried out on these devices showed that the yield strength of a plastic increases logarithmically with the rate of loading, *Fig.* 3; that an increase in temperature causes a change in the yield strength which is similar to that caused by a decrease in the rate of straining, *Fig.* 3; and that it is usually possible to find two distinct energy drops for thermoplastics in the velocity range covered, *Fig.* 4. That is to say, at these critical points, the energy required to break the test specimens of methacrylate, vinyl and ethyl cellulose showed sharp drops. With nylon, however, the energy to break showed sharp peaks at the critical velocities. Phenolic, which does not exhibit any plastic deformation even at very slow rates of straining, does not exhibit such critical points; the relation between energy to break and rate of straining is expressed in a straight line. Polystyrene behaves in the same way.

The most general, practical conclusion is that a single-velocity impact test at one temperature is not an adequate guide in the design of plastic components. When the component will be subjected in service to a wide range of temperature and rates of straining, corresponding laboratory tests must be made to obtain the necessary data or experimental parts must be tried under the expected service condition.

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1. B. Maxwell and L. F. Rahm—"Impact Testing of Plastics: Elimination of the Toss Factor," *ASTM Bulletin, No.* 161, Oct., 1949.
2. B. Maxwell, J. P. Harrington and R. E. Monica—"The Tensile Impact Properties of Some Plastics," *Princeton University Plastics Laboratory Technical Report 24A*, Jan. 31, 1952. [This research was sponsored jointly by the Army, Navy and Air Force under Signal Corps Contract No. DA-36-039-sc-133; File No. 14403-PH-51-91 (3957)].

QUALITY CONTROL METHODS

Their Use In Design

Part 5—Building-In Quality

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BOTH design and production operations benefit from statistical quality control techniques and another real advantage lies in the fact that the amount of inspection needed for the product can be dropped to a sampling basis. The customer, of course, will get a better product because quality must be built into a product—it cannot be inspected into it on a "post-mortem" basis.

Emphasis is given this factor by a case history of an automatic machine designed for a special-purpose job. It produced a high rate of defective product during a few trial runs. The designers devised a sturdier centering device and changed the slope on two cams to speed up one part of the cycle and to slow down another, *Fig.* 28. The ideas were aimed in the right direction. A few more runs showed some erratic performance but there was apparently a drop to an average of 5 per cent defective. The chief designer argued it would be uneconomical to try further refinements.

It was agreed that a maximum of 1 per cent defective was a goal that would minimize customer complaints. Attainment of this goal was to be accomplished by using statistically correct sampling tables. Picked was a sampling plan for a 1 per cent Average Outgoing Quality Limit (AOQL). Since the usual lot size was around 1000 pieces, the inspector was told to take a sample of 120 pieces and accept each

Fig. 28—Whenever a design change is being considered it would be profitable to be able to review the record of inspections from Lot Plots on file

lot if he found two or less defectives.[1]

Three or more bad pieces in the sample would require the entire lot to be screened 100 per cent. This practice resulted in a mixture of material in the storeroom made up of two types of releases. First, there would be those lots that produced samples that were accepted because two or less defective parts appeared. Then, some of the releases would not be defective at all because bad samples rejected the lot—the bad pieces had been sorted from such entire lots. The percent defective of the mixture is thus "watered" by the screened lots. The final long-run percent defective would always average out at some value equal to or less than 1 per cent. That is—this was the way it should have worked.

Inspection cost became the first problem met from the inspector's attempt to inspect quality into the outgoing product. Almost every lot failed the sampling plan and had to be screened. Everyone was hoping that the next lot sampled might somehow produce an acceptable sample to help offset the cost of screening. But according to the laws of chance, with this plan a lot 4.3 per cent defective would have just one chance in ten of passing.

Actually, more than one lot in ten did pass—but for two other reasons. The inspector unwittingly had overlooked some defective pieces in those samples. The human element in inspection had been neglected by those who decided to use attribute sampling. He only rejected one or two pieces when actually the sample of 120 sometimes had three, four, and even five bad units. Then, a very important rule was overlooked. The statistical tables work correctly only if the sample is truly a random one. Random here means each and every piece in the lot must have an equal chance to be chosen. The sample was checked by spreading the pieces out and picking pieces here and there, a practice that occasionally results in a biased or "unfair" sample. To have a sample representative of the lot, the pieces must be selected with the aid of a table of random sampling numbers. Then only the laws of chance can affect the defectiveness of the sample as compared to that of the lot. These chances are accounted for in the risk levels of the plan.

Even more oversights plagued the rejected lots that were checked 100 per cent. The outgoing material ran around 3 to 4 per cent defective, and the customers complained. The inspector was fired for being careless. The company threw out sampling, and hired a new man at a higher rate of pay and asked him to do 200 per cent inspection on each lot. Another statistical tool should have been used, one that would have helped to build quality into his product in the first place—a control chart or a Multi-Vari chart. Then inspection would have had a proper place in the sequence. It would be only to confirm or deny that the quality had been properly built into the parts. In addition, the production totals would not have to allow for a 5 per cent rate of scrap.

Human Desires of a Good Inspector: There is a natural tendency for anyone with an inspection job to be rather careful of his results. When held responsible for the parts he releases, he wants to conduct his inspection on a 100 per cent basis. Convert-

[1] References are listed at end of article.

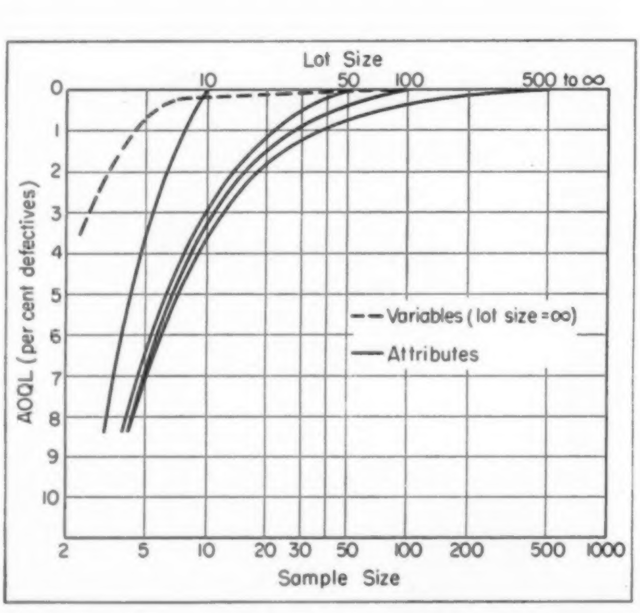

Fig. 29—Above—AOQL risk varies only with sample size for lots of 500 or more items, to infinite quantities. Sampling by variables is considerably more effective than by attributes

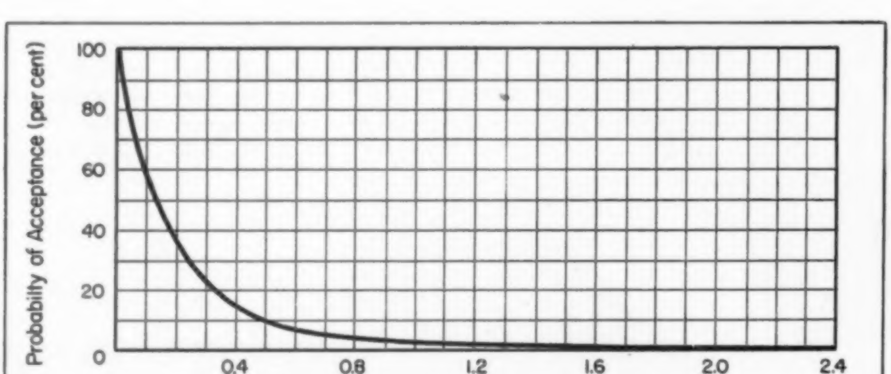

Fig. 30—The operating characteristic for the Lot Plot is the picture of its risk—how often lots of given degrees of defectiveness can be passed

ing inspectors to sampling is difficult since they know any sampling plan involves a risk—the risk of not knowing how good or bad the parts are that did not happen to fall in the sample.

In one solution the company specifies the risk levels that will be accepted. The inspectors are protected by their identification number appearing only on the parts sampled, with some other symbol not tied in to them appearing on the other parts of the lot. But also another factor is necessary—convincing the men that 100 per cent inspection is generally a dangerous practice. To demonstrate this fact, the sentence,

FEDERAL FUSES ARE THE RESULT OF YEARS OF SCIENTIFIC STUDY COMBINED WITH THE EXPERIENCE OF YEARS

can be used and, with the letter *F* considered as a defective part, individuals asked to count the number of *F*'s in that sentence. A large aircraft company located in the Midwest tried this test on over 200 individuals. Only 16 per cent got the correct answer, six. This result is unbelievable to most people. Only 80 letters make up the sentence but oversights certainly prevail for even such a small sample. The human mind is far from being as good as the human mind would like to believe.

For a final step in the conversion of an inspector, comparative tests can be made on the same lots of material by competing inspection plans to see which method overlooks fewer bad pieces. Also to be noticed is a practical element—how many inspection man-hours tie in with each method tried.

Sampling Fundamentals: In *Fig.* 29 is shown how risk varies with sample size. Using AOQL as a measure of risk serves as a common base for making a comparison among the data covered by the chart. If one or more defectives are found in any sample, that sampled lot must be completely inspected and all defective items replaced by good ones. Then in the long run the AOQL value of the chart will be the greatest percent defective that one can have in the accumulated accepted lots.

This gives a clear picture of the reduction in risk with an increase in the sample size, provided the samples are truly random and properly rejected lots are screened. Lot size has practically no effect on this relation. For a given sample size, higher risks are read from the curves for the larger lot size. But this changes quickly as the lot size increases. For all practical purposes the effect disappears for lots of 500 or more items.

There is a distinction between sampling by variables and sampling by attributes. When measurements are made on each piece, the risk becomes considerably lower for a given sample size. Go and not-go inspection takes less time per piece but pays the price of needing greater sample sizes to keep the risk at the same level as that of the variables plan.

Whenever the sample size equals the lot size this chart gives a reading of zero for the AOQL. With the lot-size curve for ten pieces this is true, since the 100 per cent inspection can be reasonably expected to have no oversights. The human failing, "inspector fatigue", sets in and causes oversights in any lot that exceeds about 50 pieces. Obviously the more pieces being checked the more routine becomes the task and

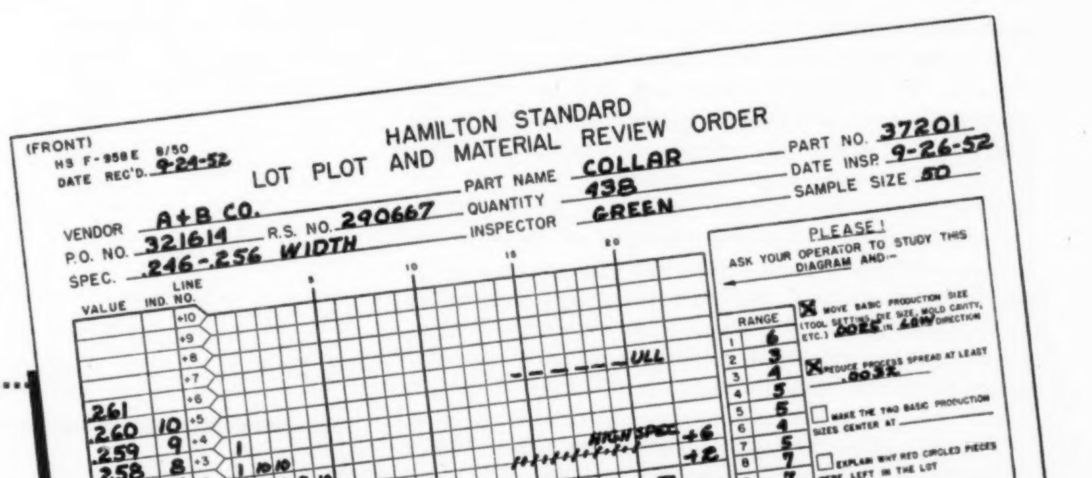

Fig. 31—Lot Plot has revealed that a shipment of collars is acceptable at the low limit but is 15 per cent defective at the high value. The worst part is more than 0.004-inch over the high limit

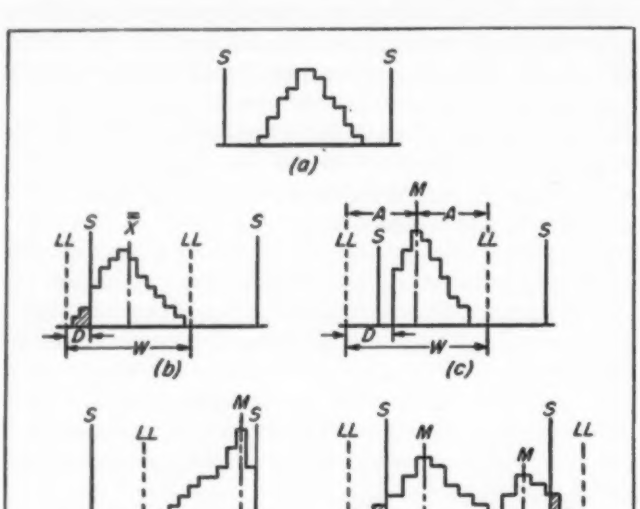

Fig. 32—The most common distributions to be found by Lot Plots. Six other types are covered in the reference of footnote 1

monotony has a more telling effect. Other factors that make the work more or less effective are: how often and how much the inspector is distracted, the type of gaging used, working conditions, and even the mental condition of the inspector. So the higher lot-size attribute curves going to zero AOQL represent a theoretical situation, virtually unattainable.

Adverse conditions are minimized when the amount of material inspected is kept low, and the inspector allowed to stop and record measurements instead of just putting work into acceptable and unacceptable piles. Unfortunately, certain characteristics can only be checked by the attribute method. Examples are the inspection of x-rays, the checking of parts for defective regions by magnetic particle or fluorescent penetrant inspection, and the use of go and not-go thread plug gages for small diameter threads (for which no successful indicator type internal diameter gages have yet been devised).

A Comparative Test of Inspection Methods: Check about 20 lots of material by all three methods—attribute sampling, Lot Plot sampling by variables, and 100 per cent inspection. Select the lots without regard to any prior knowledge of their probable defective condition. The lot size in each case should be 100 or more items. The test is run for each characteristic in the lot separately, those specifications you would now check 100 per cent. Tests for hardness are good candidates along with dimensional checks. The specification must be capable of being measured by a method that gives actual readings for each item.

For the attribute and 100 per cent checks gaging methods that are normally employed should be used. One inspector should check each lot 100 per cent exactly in accordance with present practice. Another inspector should use 500 pieces as an attribute sample. While go and not-go type of gaging will speed up this inspection, the test can be run by using a gage that gives readings if that has been the practice. However, record only whether the sampled items are found acceptable to the specification or not. A third inspector should follow the Lot Plot rules given in the following. Here he may have to use a different gaging method. Some sort of indicator set-up with gage blocks or other standards usually serves well. The indicator must be capable of being read close enough to divide the total natural variation of the characteristic found in the lot into between seven and sixteen parts.

The results of the competing methods should not be revealed to the inspectors. A supervisor should tabulate the findings recorded by each inspector. Companies that have their inspection work surveyed by government inspectors find it is generally a good idea in this test to let the government representatives in on the results. As the test progresses, rotate inspectors so that each person checks some material by all three methods. This will guarantee unbiased findings.

Conduct the 100 per cent inspection last so that any rejected items can be kept segregated for normal material review action or other such method for deciding whether they can or cannot be accepted by special deviation. Any defective items found either by the attribute sampling or by the Lot Plot sampling should be recorded by the inspector and left in the lot, since it will later be subject to 100 per cent inspection.

Table 2—Comparison of Inspection Methods

Lot Size	—Lot Plot Results— (Def. items in sample)	(% def. x lot size=def.)	Attribute Results (No. def. in sample)	100% Results (No. def.)	Several 100% Inspections (No. def.)
70	3	0.05x70=3 to 4	2	2	4
70	1	0.02x70=1 to 2	0	0	1
500	5	0.14x500=70	0	0	68
500	12	0.30x500=150	0	0	136
500	6	0.01x500=5	0	0	7
500	4	0.07x500=35	0	0	41
...	..	....	..	..	..
...	..	....	..	..	..
...	..	....	..	..	..

NOTE: A few typical entries have been made above to illustrate filling out of the form. Two lots of material were involved, one having 70 pieces and the other 500 items. In the first lot two characteristics were checked, and four were inspected in the second lot. In tabulating attribute results, of course, if the lot has 500 or fewer pieces all the material should be checked. The results entered then under both the attribute column and the 100% inspection column will be the same

It my be desirable to run another test in a few of the cases. Lots having percentages, estimated by Lot Plots, much higher than the defectives found by 100 per cent can be checked carefully for that characteristic by 200 per cent, by 300 per cent inspection, etc., to confirm the reliability of the Lot Plot. Cull out the defectives accumulated.

Tabulate results as in TABLE 2, using one line for each specification or characteristic. The purpose of these comparisons is to find the best method, disre-

garding for the moment considerations of economy. So the attribute sampling should be done at a risk level that corresponds with that of the Lot Plot plan and that comes close to the "zero" risk level of 100 per cent inspection. The sample size of 500 items with no defectives allowed is an attribute plan with almost the same operating characteristic curve as the Lot Plot has, *Fig.* 30.

The risk of a sampling plan can never be completely expressed by the use of one number. That is why some sampling tables have a numerical heading in terms of Average Outgoing Quality Limit (AOQL), others Acceptable Quality Level (AQL), and even Lot Tolerance (LTPD). And while the numbers differ, the risk may really be the same for all three. A curve is necessary to give a proper picture of the risk. An operating characteristic curve shows that when the percent defective of incoming material is zero, the probability of the sample accepting the lot is 100 per cent. But, as the material comes in more and more defective, the chance that it will be accepted drops.

AQL stands for an arbitrary, high fixed level (usually 95 per cent) of accepting lots of a low per cent defective that there would be no hesitation to use. Its value can be read from an operating characteristic curve on the horizontal scale at that point corresponding to the 95 per cent value for probability of acceptance. The Lot Tolerance is the percent defective that is generally undesirable. It is usually

Table 3—Conversion of Sums of Ranges To 3-Sigma Limits

Sums	Cells	Sums	Cells	Sums	Cells
31	4.0	47	6.1	63	8.2
32	4.2	48	6.2	64	8.3
33	4.3	49	6.4	65	8.5
34	4.4	50	6.5	66	8.6
35	4.5	51	6.6	67	8.7
36	4.7	52	6.8	68	8.8
37	4.8	53	6.9	69	9.0
38	4.9	54	7.0	70	9.1
39	5.1	55	7.1	71	9.2
40	5.2	56	7.3	72	9.4
41	5.3	57	7.4	73	9.5
42	5.5	58	7.5	74	9.6
43	5.6	59	7.7	75	9.8
44	5.7	60	7.8	76	9.9
45	5.8	61	7.9	77	10.0
46	6.0	62	8.1		

taken as the percent defective for which the probability of acceptance is rather low, say 10 per cent. As mentioned in Part 1, these fixed values of percent defective are only of theoretical interest since the material only seldom is received for sampling at just that degree defective.

The Lot Plot Plan: The results of the foregoing comparative tests are amazing to most people. The human failing to uncover defective items by 100 per cent inspection and even by attribute sampling is dramatically revealed by the Lot Plot results. While the Lot Plot falls short of being a "perfect" plan, as

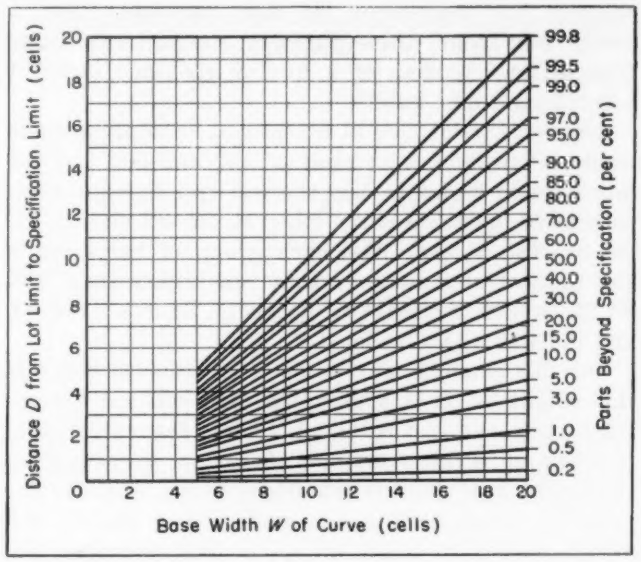

Fig. 33—A chart for finding the percentage of a normal curve that is made up of defective pieces

brought out by *Fig.* 30, it comes about as close as is practical.

The Lot Plots on file of each characteristic measured in succeeding lots have proved to be one of the valuable features of this plan. They feed back to the designer data on how poorly or how well his tolerances are being met. In many cases they not only show nonconformance but also that a simple shift in process setting will correct the trouble. Sometimes the plot shows two peaks. Here one distribution can be made of the two by paying attention to tool settings, or differences between operators. For these corrections a carbon copy of each Lot Plot becomes valuable to the supplier of the material. But also no designer should consider changing tolerances without referring to the file of these frequency distributions.

An introductory working summary, sufficient for conducting the comparative test, is about all that can be presented on the Lot Plot in this brief series.[2] The Lot Plot is a frequency histogram of special form. Lot limits come from similar but simpler steps than those used for average and range control charts. The lot limits are values where pieces can be expected to occur at the rate of from one to two per thousand.

The people who make out Lot Plots should be able to read gages, count, add, subtract, do simple multiplication, and read values from charts. The word "cell" stands for a horizontal row on the Lot Plot form where the results of the individual measurements are recorded, one entry for each piece. Any result found to be between adjacent written-in values in the left-hand column is entered in that row bounded by the two line numbers, *Fig.* 31.

To obtain a *random* sample of 50 pieces from a lot, a sampling card for that lot size is used. This card will carry 50 random numbers listed in numerical

order, none of which exceeds the lot size. The cards can be waiting on the job, having been made up in advance for convenient lot size intervals. For a handy way to get these numbers, the last two digits from a few columns of a five or six-place table of logarithms can be used.[3]

Suppose the lot size is 438 pieces. Sampling cards would be ready for lot sizes of 400 and 500 and the latter would be used. Say the last two digits of each successive log table entry gave 24, 27, 93, 08, 12, 74, 05, 31, 15, 29, etc. Use each group of three digits and discard any numbers that run higher than 500. So 242 would be used, 793 discarded, 81 used, 274, 53, and 115 also would be listed.

For this example, any number between 439 and 500 that might be on this sampling card will not be represented in the particular lot. These few extra parts to make up a full sample of 50 can be selected from the lot by any other reasonable scheme. Usually the material comes packaged in such a way that it is a simple matter to select the parts that correspond to the numbers by their position in the lot. But if the material is not packaged in orderly positions, then the entire lot should be spread out on a table having lined areas. Each piece on the table can then be "numbered" from its relative position in one of the areas.

Divide the sample into ten separate lots of five pieces each. Enter the measurements of the first five pieces by writing the number 1 in the proper squares. Each is in the cell that is bounded by the same two lines that on the indicator came above and below the actual position of the pointer. Record the number of the cell units between the lowest and the highest l's in the RANGE column of the Lot Plot. Start each count with "zero", "one", "two", etc. *Fig.* 31 shows a range of six cells for the first subgroup of five pieces. Measure the next five parts and place the figure 2 in the appropriate squares. Move one square to the right if the first square is already occupied. Record the range again, but against subgroup number 2. Continue until the ten subgroups have been entered.

Before making all these entries a scale of sizes must be determined. Measure the first five pieces and determine the average reading and the range or difference between the largest and smallest measurements. Write in the average size in the VALUE column on the form in or within a few lines of the central horizontal space with the printed 0. This will keep the entire histogram centered on the paper. Double the range of these first five readings and see if that falls between 7 and 16 intervals that you might want to use in the VALUE column.

In *Fig.* 32 the range of the first five pieces is six cells, each being 0.001-inch, giving 0.006. Double this range is 0.012 or 12 intervals, satisfactorily between 7 and 16. If this were not the case a larger or smaller interval would have to be picked to give the right sensitivity to the histogram. The doubling gives an estimate of the base width of the completed distribution. The gage used must be capable of being read to each of the intervals.

Write in measurements in these increments in the VALUE column to serve as reference scale for future plotting of results. The column marked IND. stands for actual indicator, micrometer, or other gage readings, if they differ from those printed in the next column headed LINE NO. In the pictured example the 0 for the gage happens to correspond to 0.250 or line number -5. These indicator reading entries are needed only when the line number printed figures do not serve to tell handily which cell should be used to record each measurement.

Add the ten entries in the range column and convert this sum to 3σ from TABLE III. These numbers will position the lot limits as that number of cell units above and below the average of the whole distribution.

Using a strip of printed LINE numbers cut out from a blank Lot Plot form, count out the position of the average of the entire distribution $(\overline{\overline{X}})$ as follows:

1. Locate the strip along the base of the distribution so that the +'s and −'s of the strip split the distribution about equally to the eye. The 0 of the strip should fall directly over a row of readings, rather than being offset half a cell width as is the case on the printed Lot Plot form. Over to the right of this cell write in a 0, as shown in *Fig.* 31.
2. Balance out as many minus values in a row against its corresponding plus row by moving the strip to the right. These sets of pairs of covered readings can be considered as cancelled out. Record at the right-hand section of the form, above and below the written-in 0, each excess as a plus or minus product of the strip cell number and the number of excess readings in the row. For example, in the $+1$ row there are seven entries and in the -1 row there are eight entries. The excess of one entry times -1 gives a write-in value of -1. Comparing the $+2$ and the -2 cells we have an excess of one entry in the $+2$ cell, which gives another write-in value of $+2$, and so on.
3. Total the recorded numbers, paying attention to sign, and divide by 50 to get $\overline{\overline{X}}$. This is done by simply doubling the total $(+7)$ and marking off two decimal places to the left. Then draw the $\overline{\overline{X}}$ line on the chart, moving up or down as necessary from the center of the cell that was chosen for the written-in 0. In *Fig.* 31 this is $+0.14$ cells above the approximate center of the distribution, or very close to the center of that cell.

By counting up and down from this $\overline{\overline{X}}$ line the number of cells previously found as the 3σ value, the probable limits of the lot can be shown. These should be labelled ULL (upper lot limit) and LLL (lower lot limit). In the example they are located at 6.6 cells above and below the average line. Mark in the specification limits on the form to complete the job.

How To Analyze Lot Plots: The picture obtained by the foregoing plan should be compared with one of the five types of Lot Plots shown in *Fig.* 32. Type *a* is a normal-shaped, symmetrical distribution that lies well within the specification limits. Even the lot limits do not need to be found. The lot can be considered acceptable for this characteristic.

Type *b* is also a normal-shaped, symmetrical dis-

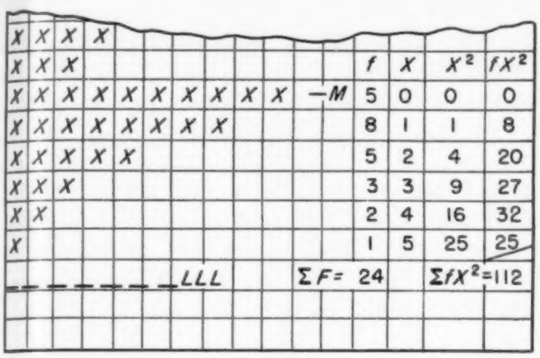

Fig. 34—Schematic section of method used for handling skewed or unbalanced distributions

ribution but a portion of it extends beyond a specification limit. By counting the number of cells in the distance D and in the base width distance W and entering these results in the chart of *Fig.* 33, from the right-hand side of the labelled slanting lines can be read the percentage of the entire lot that can be expected to be beyond that specification. This Lot Plot can be then reviewed for disposition as accept, return to source, screen, rework, or scrap.

Type *c* is a distribution with one tail cut off. The lot limit for the complete half of the distribution that has a filled-out tail should be computed in accordance with the sectional illustration of *Fig.* 34. The peak of the distribution or the cell with the greatest number of readings is labelled M for mode. Under f are tabualted the number of readings in each cell from the mode cell out to the end of the tail, but only one-half of the entries in the mode cell are used. This column is added to obtain ΣF. In the next column labelled X, the mode cell is called 0 and each cell numbered in sequence out to the tail. In the next column enter the square of these cell numbers. Finally write the product of the f value and the X^2 value. Add this last column to give ΣfX^2. Enter these two sums in *Fig.* 35 or 36, whichever has the appropriate range of values, to read as an answer on the bottom of the chart the location of the lot limit from the mode in cells.

This number of cells represents distance A of the type *c* Lot Plot. The two lot limits, set at equal distances away from the mode, stand for a prediction of the extreme pieces of the distribution if it had not been cut off. When these lot limits are within specification limits, material should be accepted with this characteristic. If the lot limit at the filled-out tail is outside of specification limits, the percentage of the entire lot that is defective should be predicted by following these steps:

1. Count the number of cells from the center of the mode cell to this lot limit. In the cut-away of Figure 34 it is 6.4 cells. Double it, calling the 12.8 cells W
2. Count the number of cells that the lot limit is beyond the specification limit. Let us say this might be 4.9 cells. Call this distant D
3. From D and W read off from *Fig.* 33 that the lot would be 25 per cent defective if it had filled out into a complete normal curve
4. Multiply ΣF by 0.04. In this example it is 0.04 times 24 = 96
5. Multiply the *Fig.* 33 percentage by this factor, 0.96 x 25 per cent = 24 per cent. This value is marked in the appropriate block at the bottom center of the Lot Plot form as being the percentage of the lot that is beyond the specification in question.

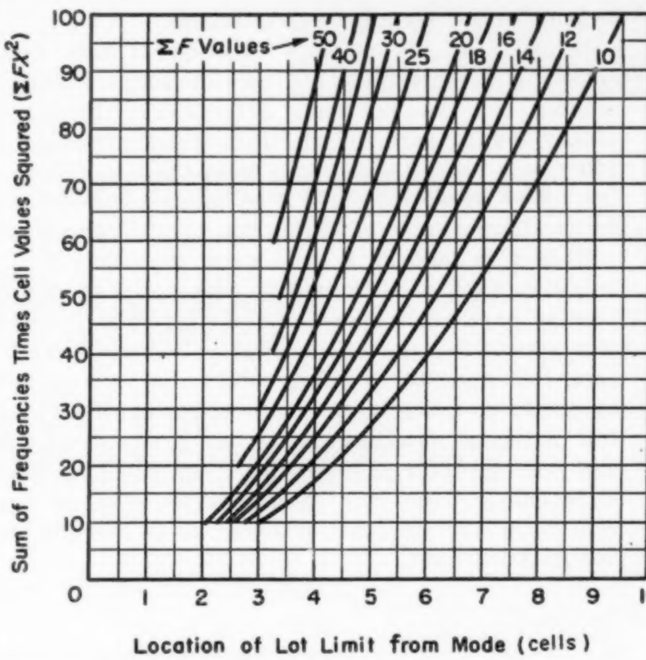

Fig. 35—Chart for finding lot limits for ΣfX^2 values less than 100

If the lot limit at the cut-off side of the distribution lies beyond the specification limit, it is not certain how much of the distribution may or may not be defective. It is known that it is no greater than that found if the tail filled out. So again with the count of distance D on the cut-off side and distance W, which is already known, the maximum percentage can be read directly from *Fig.* 33 without finding an adjustment factor.

When this percentage and the extent of the error as indicated by the position of the lot limit are acceptable on a material review basis, disposition of the lot can be made. If not acceptable, this characteristic has to be checked by an attribute sampling plan at this specification limit. When the main function of the characteristic being plotted is to result in a certain design fit or clearance with a mating part, as a shaft or bore diameter or a thread pitch diameter, etc., 25 additional pieces must be taken at random from the lot. If none lie beyond this spe-

cification limit (as well as none of the lot plotted pieces falling beyond it) enter the results in the bottom right-hand section of the Lot Plot form and accept the lot for this characteristic.

However, if the primary function of this characteristic is structural, as a wall thickness, hardness reading, etc., thus not being influenced by the chance of any mating part, an additional random sample of 250 pieces needs to be taken from the lot. Consider the lot as acceptable only if no parts are found defective from sample of 300 at that specification limit.

Type *d* is half or a little more than half a distribution, such as is formed by concentricity, squareness or parallelism specifications. The 0 at the bottom stands for the cell boundary corresponding to zero full indicator reading while the tail extends on out to so many tenths or thousandths of an inch, full indicator reading. Again mark the mode cell and use the method of analysis described for type *c*, including the factor to modify the ***Fig.*** 33 percentage if the lot limit is found to be beyond the specification limit.

In type *E* two or more distributions are indicated. An additional sample can be taken, if necessary, or the cell interval distance changed where required to give at least 25 readings from each external mode out toward the tail, if it is expected that a lot limit will be found to be beyond a specification limit. If not, no lot limits need to be computed and the characteristic can be considered acceptable. When computing lot limits, the method described for type *c*, for locating the lot limits from the modes that are closest to the specification limits, should be used along with the factor for modifying the ***Fig.*** 33 percentage at each end that might be defective.

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2. For a complete description of the Lot Plot that tells how to handle stray pieces, how to use statistical rules for material review dispositions, etc., the reader is referred to *Industrial Quality Control*, "The Hamilton Standard Lot Plot Method of Acceptance Sampling by Variables", July 1950, and "Recent Lot Plot Experiences Around the Country", March 1952.
3. Although this is a practical source of such numbers, a more correct source (tested by several means for randomness) is *Tables of Random Sampling Numbers*—M. G. Kendall and B. Babington Smith, Cambridge University Press, 51 Madison Avenue, New York, N. Y.

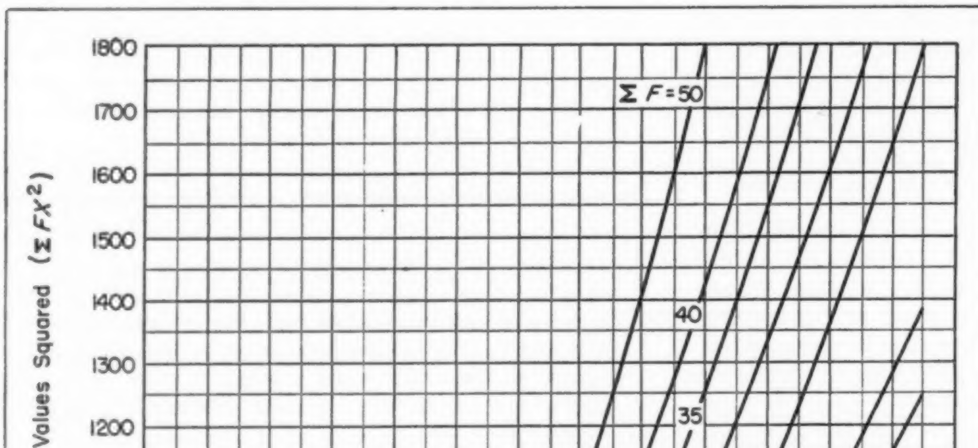

Fig. 36—Complementary chart for determining lot limits for ΣfX^2 values over 100

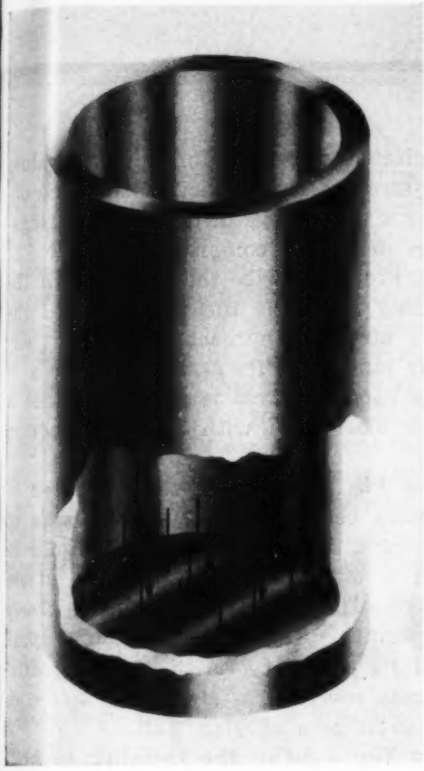

Fig. 1 — Sketch showing case of axially loaded container with flat end plate

STRESS AND DEFLECTIONS IN
Axially Loaded Cylindrical Containers

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STRESSES and deflections of a cylindrical container and its end plate are given in this article for the case of uniform loading applied to the end plate, *Fig.* 1. This type of loading is met approximately in practice when a cylindrical container filled with a nonfluid, nonstructural material is subjected to gravitational loading in the axial direction, such as from the weight of the material or from a vertical shock. A typical example might be an upright cylindrical container loaded with sand or coal. The radial loading applied to the container by materials of this type is small relative to the axial loading.

Method of Solution: The method of analysis employed in arriving at working equations consisted of considering the end plate and the cylinder separately. The general plate and cylinder differential equations given by Timoshenko[1] were solved for this particular case. The arbitrary constants appearing in the solution of the equations were evaluated from the boundary conditions of this particular problem; namely, that the conditions of equilibrium must hold at the junction of the plate and cylinder. That is, the moment on the plate must equal the moment on the cylinder, the slope of the plate edge must equal the slope of the cylinder, etc.

The assumptions made in this analysis are the usual ones of elastic theory: plate deflections due to bending are small compared to the plate thickness; the material is perfectly elastic, homogeneous and isotopic; plate loading is uniform; radial loading on the cylinder is negligible; and plate thickness, cylinder wall thickness and cylinder displacement are small compared to the plate diameter.

[1] References are tabulated at end of article.

Stress and Deflection Equations: In the original analysis[2] equations were developed for the stresses and deflections at any point on the plate or cylinder. Here, however, only the maximum stresses and deflections are given since these values are the limiting conditions in practical design.

The results are given in generalized dimensionless equations in the accompanying tabulation. Curves are plotted from these expressions for the specific case of Poisson's ratio of 0.3 in *Figs.* 2 to 8. Stresses and

Nomenclature

a = Plate radius, inches
E = Modulus of elasticity, psi
h = Plate thickness, inches
L = Cylinder length, inches
q = Uniform plate loading, psi
t = Cylinder wall thickness, inches
W_B = Axial bending displacement in plate, inches
W_T = Cylinder tensile elongation, inches
$= Laq/2tE$
$\alpha = t/h$
$\beta = a/h$
γ = Reciprocal of cylinder "wave length," inches^{-1}
$= \sqrt{3(1-\mu^2)}/\sqrt{ah}$
δ = Cylinder radial displacement, inches
μ = Poisson's ratio
σ_r = Radial plate bending stress, psi
σ_t = Tangential plate bending stress, psi
σ_y = Cylinder axial stress, psi

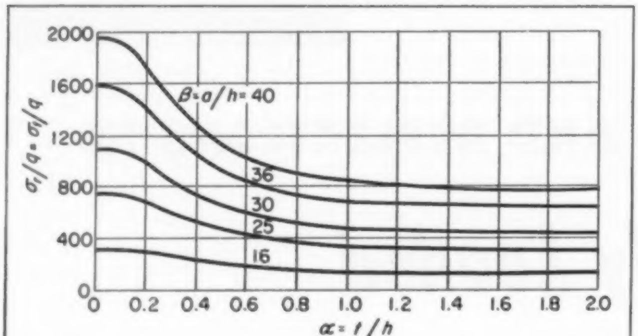


Fig. 2—Radial plate bending stress at periphery of plate

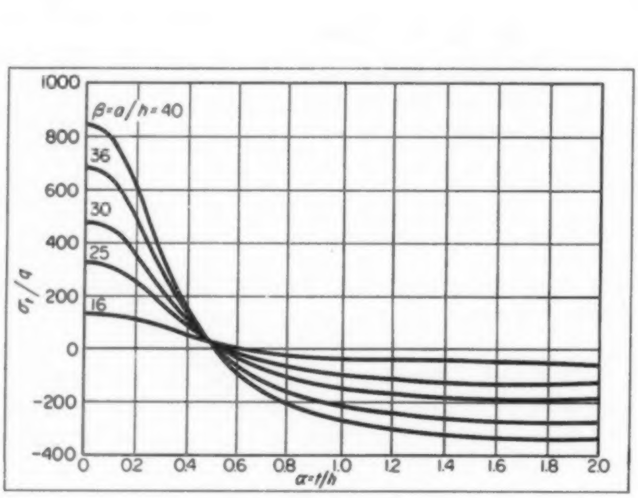


Fig. 3—Tangential plate bending stress at edge of plate

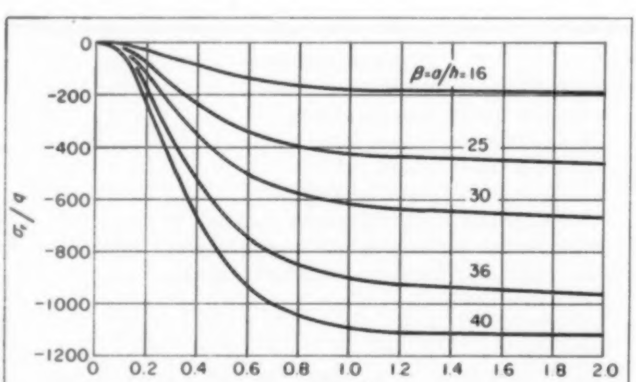


Fig. 4—Radial and tangential plate bending stresses at center of plate

deflections are plotted versus α for several values of β so that the stresses and deflections may be obtained directly for a wide range of dimensions. Negative stress values designate compression.

Equation 1 and *Fig.* 2 describe the variation of the radial plate bending stress at the periphery of the plate. For $\alpha > 1.2$ stresses approach those of a circular plate rigidly built-in at its periphery. For $\alpha = 0$ the radial plate bending stress is zero since this is the case of a circular plate with freely supported edges.

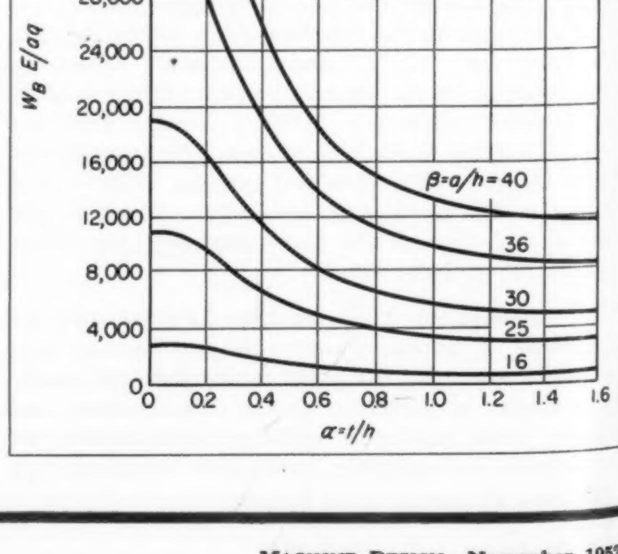
Equation 2 and *Fig.* 3 illustrate the variation of the tangential plate bending stress at the edge of the plate with α for different values of β. Again as α increases from zero to 1.2 or greater, the stress conditions vary from those of a circular plate freely supported at its periphery to those of a circular plate rigidly built-in at its periphery. Also, the tangential plate bending stress reverses in sign as α increases and this stress equals zero at $\alpha \approx 0.55$.

Equation 3 and *Fig.* 4 define the variation of both the radial and tangential plate bending stress at the center of the plate with α. At the center of the plate the radial and tangential bending stresses are equal.

Equation 4 and *Fig.* 5 define the maximum axial plate displacement due to bending. To obtain the total axial displacement, the cylinder elongation, $Laq/2tE$, must be added.

Equation 5 and *Fig.* 6 illustrate the variation of

Fig. 5—Maximum plate displacement due to bending



the maximum axial cylinder bending stress with α. This stress peaks at $\alpha \approx 0.2$. Obviously, values of α in this region are to be avoided.

Equation 6 and *Fig.* 7 illustrate the variation of the maximum cylinder axial stress with α. This stress differs from that of Equation 5 and *Fig.* 6 in that the tensile stress in the cylinder has been superimposed on the bending stress, giving the total maximum stress on the cylinder. The curves of *Fig.* 7 differ from those of *Fig.* 6 in that the curves of *Fig.* 6 went to zero for $\alpha = 0$ whereas the curves of *Fig.* 7 rise asymptotically with the vertical axis at $\alpha = 0$. As the cylinder thickness decreases, the tensile stress increases rapidly so that the total stress approaches infinity even though the bending stress goes to zero. Again, the curves peak at $\alpha \approx 0.2$, an undesirable range for the ratio of cylinder wall thickness to plate radius.

Equation 7 and *Fig.* 8 describe the variation of the maximum cylinder displacement in the radial direction with α. Two vertical axes are shown to cover different ranges of cylinder displacement since this displacement decreases rapidly with α.

Example: Assume a vertical cylindrical container with a flat end plate having the following dimensions to be loaded with 6000 lb of powdery material: cylinder radius, $a = 25$ inches; cylinder wall thickness, $t = \frac{1}{2}$-inch; end plate thickness, $h = \frac{7}{8}$-inch; and cylinder length, $L = 100$ inches. Assume, also, that the container is steel, with Poisson's ratio, $\mu = 0.3$ and modulus of elasticity, $E = 30{,}000{,}000$ psi. Since the container is loaded with nonfluid material, the radial loading can be neglected.

First, the unit loading applied to the end plate is $q = 6000/2\pi(25)^2 = 1.52$ psi. Then, the dimensionless ratios, α and β, are $\alpha = t/h = 0.5/0.875 = 0.572$ and $\beta = a/h = 25/0.875 = 28.7$.

Stress and Deflection Equations

Radial plate bending stress at periphery of plate:

$$\frac{\sigma_r}{q} = \frac{3}{4}\beta^2\left(\frac{1+\mu}{2\sqrt[4]{3(1-\mu^2)}\sqrt{\beta}\,\alpha^{2.5}+1+\mu} - 1\right) \qquad (1)$$

Tangential plate bending stress at periphery of plate:

$$\frac{\sigma_t}{q} = \frac{3}{4}\beta^2\left(\frac{1+\mu}{2\sqrt[4]{3(1-\mu^2)}\sqrt{\beta}\,\alpha^{2.5}+1+\mu} - \mu\right) \qquad (2)$$

Radial and tangential bending stresses at center of plate:

$$\frac{\sigma_t}{q} = \frac{\sigma_r}{q} = \frac{3}{8}\beta^2(1+\mu)\left(\frac{2}{2\sqrt[4]{3(1-\mu^2)}\sqrt{\beta}\,\alpha^{2.5}+1+\mu} + 1\right) \qquad (3)$$

Plate displacement due to bending:

$$\frac{W_B E}{a q} = \frac{3}{16}\beta^3(1-\mu^2)\left(\frac{4}{2\sqrt[4]{3(1-\mu^2)}\sqrt{\beta}\,\alpha^{2.5}+1+\mu} + 1\right) \qquad (4)$$

Maximum cylinder axial bending stress:

$$\frac{\sigma_{yb}}{q} = \frac{1.5\sqrt[4]{3(1-\mu^2)}\sqrt{\alpha}\,\beta^{2.5}}{2\sqrt[4]{3(1-\mu^2)}\sqrt{\beta}\,\alpha^{2.5}+1+\mu} \qquad (5)$$

Maximum cylinder axial stress:

$$\frac{\sigma_{y\text{-}max}}{q} = \frac{1.5\sqrt[4]{3(1-\mu^2)}\sqrt{\alpha}\,\beta^{2.5}}{2\sqrt[4]{3(1-\mu^2)}\sqrt{\beta}\,\alpha^{2.5}+1+\mu} + \frac{\beta}{2\alpha} \qquad (6)$$

Maximum cylinder radial displacement:

$$\delta\gamma\frac{E}{q} = \frac{0.485(1-\mu^2)\beta^3}{2\sqrt[4]{3(1-\mu^2)}\sqrt{\beta}\,\alpha^{2.5}+1+\mu} \qquad (7)$$

Data Sheet

STRESSES: From *Fig.* 1 for $\alpha = 0.572$ and $\beta = 28.7$, $\sigma_r/q = 450$ or $\sigma_r = (450)\ (1.52) = 685$ psi.

From *Fig.* 2 for the prescribed values of α and β, $\sigma_t/q = 0$ or $\sigma_t = 0$ psi.

From *Fig.* 3, $\sigma_r/q = \sigma_t/q = 560$ or $\sigma_r = \sigma_t = (560)\ (1.52) = 851$ psi.

From *Fig.* 5, $\sigma_{y-max}/q = 1400$ or $\sigma_{y-max} = (1400)\ (1.52) = 2180$ psi.

DISPLACEMENTS: Maximum plate displacement in the axial direction is made up of two components. One is the deflection of the plate center due purely to plate bending. The other component of plate displacement is the tensile elongation of the cylinder. The first component is found from *Fig.* 4: $W_B\ E/aq = 7600$, or $W_B = 7600qa/E = (7600)\ (1.52)\ (25)/30{,}000{,}000 = 0.00962$-inch.

The cylinder tensile elongation is $W_T = Laq/2tE = (100)(25)(1.52)/(2)(0.5)(30{,}000{,}000) = 0.000126$-inch. Total axial displacement is $W_B + W_T = 0.009746$-inch. For maximum cylinder radial displacement, from *Fig.* 7, $\delta\gamma E/q = 1800$. But

$$\gamma = \sqrt{\frac{3(1-\mu^2)}{a\,h}} = \sqrt{\frac{3(1-0.09)}{(25)\,(0.875)}}$$

$$= 0.354\text{-inch}$$

Then, $\delta = 1800q/E\gamma = (1800)(1.52)/(30{,}000{,}000)(0.354) = 0.000258$-inch.

SHOCK LOADING: The maximum stress for this example is the cylinder axial stress, 2180 psi, and the maximum displacement is the total plate displacement, 0.00975-inch. Stress and deflection are rather small. However, had a shock been applied to the container, the loading would have been multiplied by the g shock acceleration. For example, a $5g$ shock would have imposed five times as much loading on the cylinder, resulting in much larger stresses and deflections.

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2. D. P. Timo—*Axial Loading of a Cylindrical Container*, Report No. 495, Knolls Atomic Power Laboratory.

Fig. 6—Maximum axial cylinder bending stress

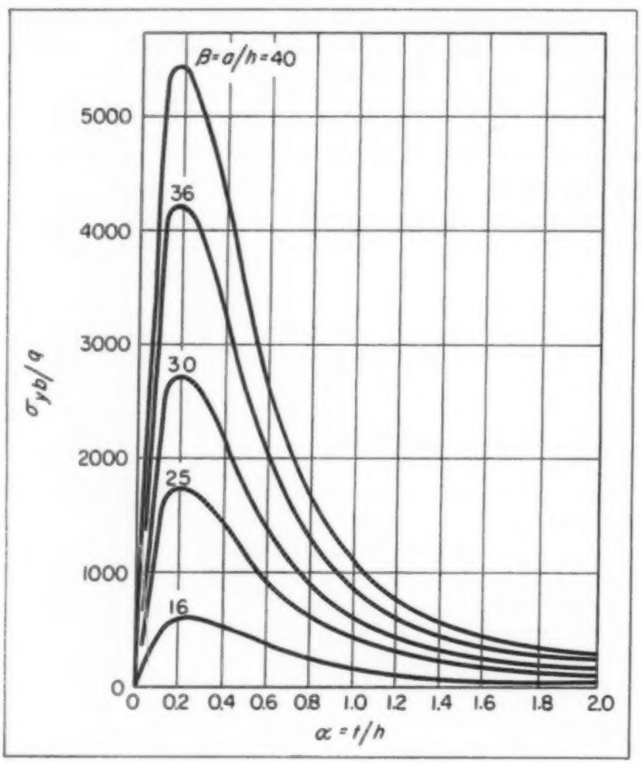


Fig. 7—Maximum cylinder axial stress

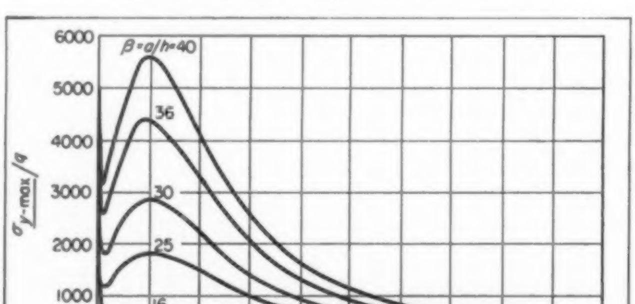
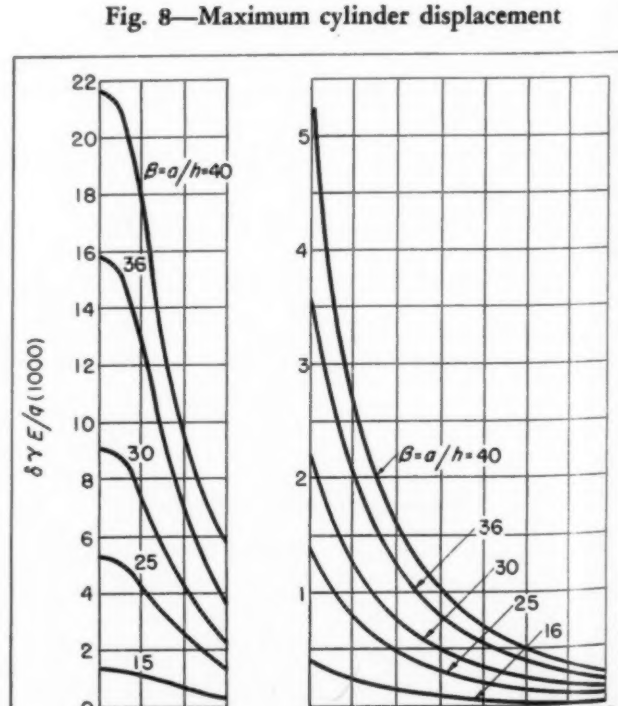


Fig. 8—Maximum cylinder displacement



Analyzing Regulator Performance

. . . with analog computers simplifies design procedures

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PERHAPS the most important factor contributing to the highly perfected regulating systems of today is the application of advanced analytical techniques to their design. Of particular significance is the use of analog computers to determine the performance of new circuits before they are built.

In any regulating system, many variables affecting system operation are involved. Widely differing mechanical and electrical functions must be made to operate with very precise relationships to each other. To determine, in advance, the performance of such a system under all conditions would be impossible without computers.

Analysis of regulating systems consists of three major steps:

1. General layout of the regulator and analysis of the performance of component parts.
2. Translation of the component-part performance into overall-system performance under specified conditions.
3. Interpretation of overall-system performance in terms of end product.

Prior to the development of the analog and other suitable computers, by far the most difficult phase was the second requirement — translation of component-part per-

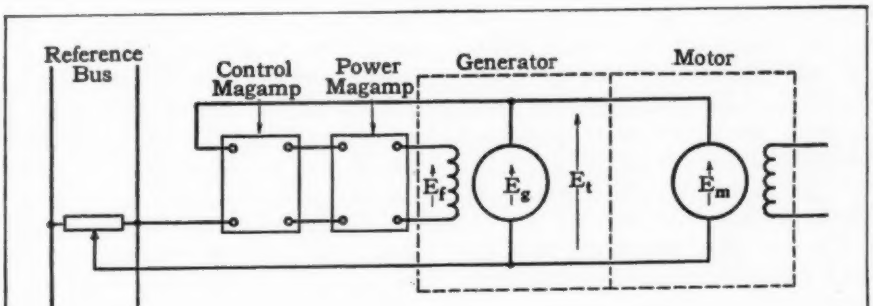
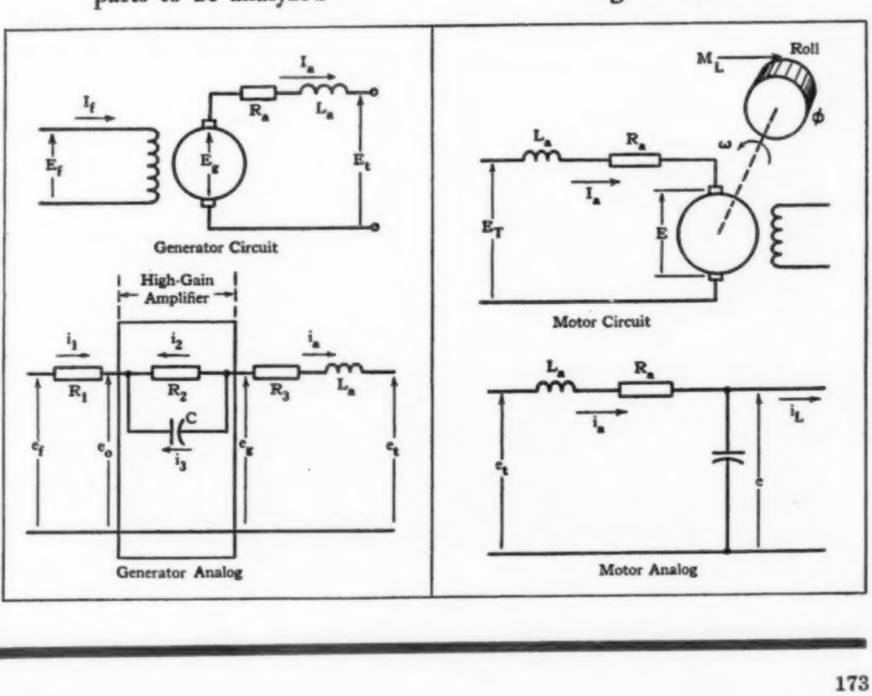


Fig. 1—Above—General layout of regulating system for cold strip tandem mill showing component parts to be analyzed

Fig. 2—Below—From breakdown of system in *Fig.* 1 components can be reduced to simplified electrical analogs as shown



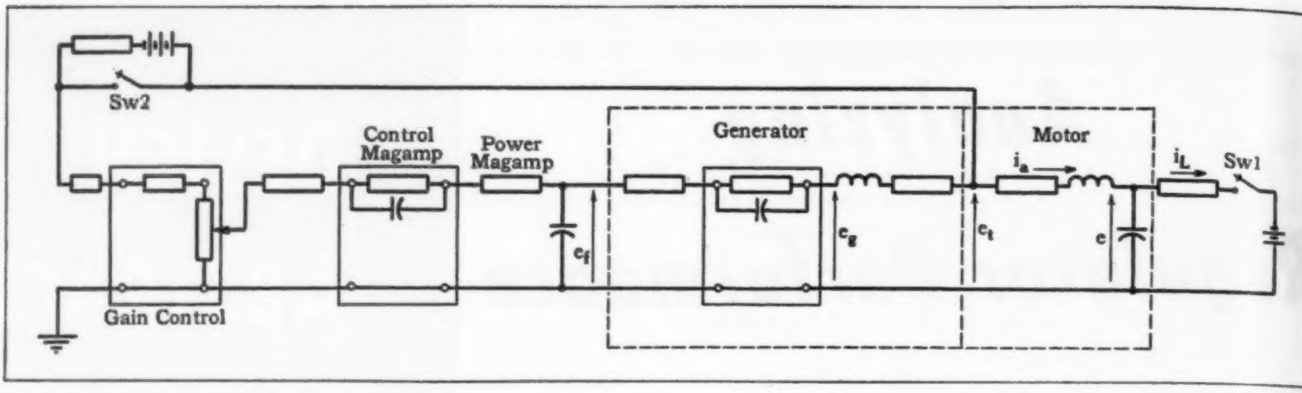


Fig. 3—Completely developed analog for analysis of regulator system in *Fig.* 1

formance into overall-system performance. To make the problem amenable to solution, simplifying assumptions were necessarily made, such as linearization of component characteristics, and some factors were assumed to be negligible simply because they could not be coped with by usual methods of calculation. As a result, the designer could determine fairly well the steady-state performance, but could predict only sketchily the dynamic performance of his regulating system.

The analog computer has reduced this heretofore difficult phase of analysis to a rather simple and rapidly executed procedure. For example, consider the regulating system for a cold strip tandem mill in which the general layout of the system and the analysis of component parts has been completed and is as shown in the diagram in *Fig.* 1.

With the analog computer all components and factors, whether electrical or mechanical, are reduced to simplified electrical analogs. Therefore, the first step is to devise the electrical analog of the system. The procedure for doing this, by components, is shown in *Fig.* 2. First, the analog for the motor is devised from several relationships which can be immediately set up. Motor speed can be related to induced emf; motor torque can be equated to a function of armature current or written in terms of load and acceleration; and terminal voltage can be expressed in terms of induced emf and armature current. Combining these several relationships, an equation between terminal voltage, motor speed, and load torque can be established in terms of mechanical time delay and armature time constant as

$$E_t = G\left[\omega + T_m\frac{d}{dt}\left(\omega + T_a\frac{d\omega}{dt}\right)\right] + \frac{R_a}{B}\left(M_L + T_a\frac{dM_L}{dt}\right)$$

Symbols are defined in the Nomenclature and *Figs.* 1, 2, and 3.

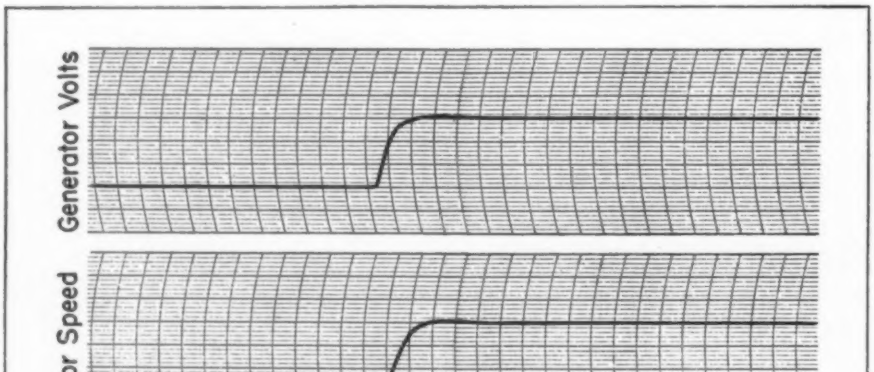
Nomenclature

A = Amplifier gain
B, C, G, K = Constants
E = Induced emf, motor
E_f = Field voltage, generator
E_g = Internal voltage, generator
E_t = Terminal voltage
I_a = Armature current
I_f = Field current, generator
L_a = Armature inductance
L_f = Field inductance, generator
M = Motor torque
M_L = Load torque
R_a = Armature resistance
R_1, R_2, R_3 = Resistances, generator analog
R_f = Field resistance, generator
$T_a = L_a/R_a$ = Armature time delay
$T_m = R_aC$ = Mechanical time delay, motor
$T_g = R_a\phi/BG$ = Mechanical time delay, generator
ω = Angular velocity
ϕ = Inertia

In a computer operation such as this, electrical and mechanical quantities must be combined into one electrical analog. In the motor, speed and voltage are equivalent, as are current and torque, and inertia can be represented by a

(*Continued on Page 260*)

Fig. 4—Response of regulating system to suddenly applied change in voltage setting. Other changes in operating conditions may also be easily simulated and recorded for design analysis



NEW PARTS

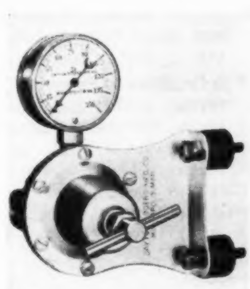
AND MATERIALS

... presented in quick-reference data sheet form for the convenience of the reader. For additional information on these new developments, see Page 193

PRESSURE REDUCING REGULATOR — 1

... regulates pressures to 150 lb.

Dayton Rogers Mfg. Co., Minneapolis 7, Minn.



Vibration mountings eliminate vibration, shock and concussion of the regulator.

Size: ¼, ⅜, ½ and ¾-in. pipe size connections.

Service: Reducing, regulating and gaging compressed air pressures to 150 psi.

Design: Combination pressure-reducing regulating valve and gage with mounting bracket and vibration dampening mounts; gage valve adjusting screw has locknut; available without gage.

For more data circle MD 1, Page 193

GEAR OIL — 2

... contains molybdenum disulfide

Imperial Oil and Grease Co. Inc., 6399 Wilshire Blvd., Los Angeles 48, Calif.

A metallic film is formed which reduces friction and wear.

Form: Fluid lubricating oil.

Size: 115 lb and 55 gal drums.

Service: Lubricating gears and bearings; molybdenum disulfide film will withstand surface stresses of 108,000 to 225,000 psi; effectively fills "valleys" or pores of metal surfaces to form metallic film which reduces friction and wear.

Properties: Molybdenum disulfide particles, less than micron size, suspended in paraffin-base mineral oil; wear reduction is due to structure of molybdenum disulfide particle which is composed of sheet of molybdenum atoms "sandwiched" between two sheets of sulfur atoms; one sheet of sulfur adheres very strongly to gear or bearing due to strong metal-sulfur bond while reverse side slips due to lack of bonding to sulfur layer of other particles; available in all SAE viscosity grades.

For more data circle MD 2, Page 193

SINTERED NYLON — 3

... basically similar to molded grades

National Polymer Products Inc., Box 422, Reading, Pa.

Produced by powder metallurgy techniques, parts from this material have similar hardness, chemical and wear resistance as standard nylon.

Designation: Nylasint 66.

Form: Powder, with particle size below 10 microns.

Service: For cold-pressed and sintered bearings, gears, cam rollers, valve seats; parts have hardness, chemical and wear resistance comparable to Nylon FM 10001 but somewhat lower toughness than molded product; good dimensional stability, since processing below melting point causes less internal strain; can be filled with lead, copper, graphite, refractory oxides, magnetic materials, etc., for special properties; low coefficient of friction.

Properties: Nylon powder; ultimate particle size, 1-3 microns; density, cold pressed before sintering, 1.08 gm per cc.

For more data circle MD 3, Page 193

CHECK VALVE — 4

... has no wearing parts

Techno Co., 28 W. 12th St., Erie, Pa.

No grinding or replacing of valve seats is necessary.

Size: 4 to 16 in.

Service: Preventing reverse flow in air or noncorrosive gas lines at pressures to 50 psi and temperatures to 200 F; pressure drop of ½-in. of water at 2000 cfm and 25 psi discharge pressure.

Design: Check valve consists of steel body with semi-flexible, "feathering" valve element; specials available for higher than standard pressures and temperatures; special materials and linings also available.

For more data circle MD 4, Page 193

NEW PARTS AND MATERIALS

PACKING RINGS 5
. . . molded endlessly, with no joint

Periflex Inc., Hazel Park, Mich.

Rings are available for use with machined metal end members, or can be supplied with end adaptors.

Designation: 88.

Size: ¼, ⁵⁄₁₆ and ⅜-in. cross-section; ½ to 3 in. ID, 1 to 3½ in. OD.

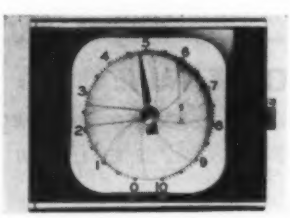
Service: Rod or piston sealing; seals pressures over 5000 psi; withstands high temperatures and flexing.

Design: Endlessly molded packing ring sets; molded from special neoprene compound; cross-section of rings is such that higher pressures increase sealing action; lip interference of rings automatically preloads packing in stuffing box, eliminating need for any but initial gland adjustment; channel formed between adjacent rings acts as lubricant reservoir.

For more data circle MD 5, Page 193

ROUND-CHART RECORDER 6
. . . is moisture, fume and dustproof

Bristol Co., Waterbury 20, Conn.



Can be used with any sensing element whose dc voltage or current output changes as a function of the measured variable.

Designation: Dynamaster.

Size: 19 in. wide, 15½ in. high, 13⁵⁄₁₆ in. deep; chart, 12 in. diam.

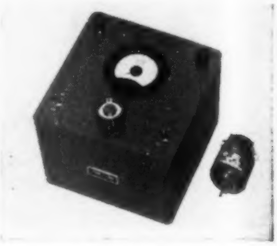
Service: For measuring and recording any quantity that can be measured in terms of direct voltage or current; as many as 16 variables can be measured and recorded by one recorder; operates on 120 v, 60, 50 or 25 cycle ac; 24-hour and 7-day chart speeds; 20, 7, 3, 1.5 and 2/3-sec full-scale pen speeds.

Design: Continuous-balance null-type potentiometer; variation in measured quantity is electronically amplified to direct operation of balancing motor which rebalances potentiometer by moving slidewire contact; motor also drives recording and indicating mechanisms, controlling models for air or electrical control also available.

For more data circle MD 6, Page 193

MOTOR AND CONTROL 7
. . . give speeds to 10,000 rpm

Industrial Control Co., Wyandanch, Long Island, N. Y.



Meter on front panel of control box permits direct reading of motor speed.

Designation: 302-A.

Size: Control box—8 in. square, 6½ in. deep; motor—2¼ in. diam, 3 in. long.

Service: Supplying a maximum of 1 oz-in. torque at speeds from 100 to 10,000 rpm; operates from 117 v 60-cycle ac line; speed may be varied remotely, if desired.

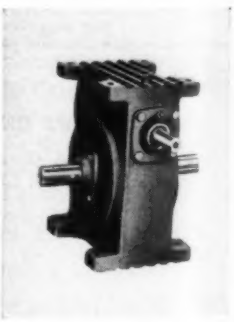
Design: Small, high-speed motor and control; matching gear boxes available in any desired ratio; provision for measuring torque delivered to load.

Application: Servo directors, automatic control systems; light machine tools; industrial process controls; torque measurement.

For more data circle MD 7, Page 193

SPEED REDUCER 8
. . . can be mounted in three positions

Eberhardt-Denver Co. 1408 W. Colfax Ave., Denver, Colo.



Each speed reducer is operated under load and tested before being shipped.

Designation: 3L2.

Size: ½ to 3 hp; 3½ in. from center of input shaft to center of output shaft.

Service: Speed reducing, with 9 ratios available; may be mounted horizontally with output shaft above or below input shaft or with output shaft vertical.

Design: Worm gear speed reducer; integral worm and input shaft of hardened ground and polished alloy steel; output shaft of ground, high-carbon steel; worm gear is high grade bronze; both shafts mounted in Timken bearings; top and bottom surfaces of housing machined so that either may be used as mounting base.

For more data circle MD 8, Page 193

NEW PARTS AND MATERIALS

ILLUMINATED PANEL METERS 9

. . . combine small size and accuracy

International Instruments Inc., P.O. Box 2954, New Haven 15, Conn.

Externally mounted miniature aircraft lamp illuminates translucent plastic scale through small sealed window in back of meter case.

Designation and Size: 150 is 1.688 in. diam, mounts in 1 17/32 in. diam hole, weighs 1½ oz; 153 is 1.750 in. square, mounts in 1 17/32 diam hole, weighs 1¾ oz.

Service: Dc microammeters, 0-100 to 0-500; dc milliammeters, 0-1 to 0-500; dc ammeters 0-1 to 0-15; dc millivoltmeter, 0-50; dc voltmeters, 0-1 to 0-500, resistance, 1000 ohm per v; ac voltmeters, 0-1 to 0-500; accuracy ±3% on dc meters, ±5% on ac meters.

Design: Miniature D'Arsonval type movements in black, anodized aluminum cases; 1.322 in. scale length; left-hand zero; translucent scale illuminated by externally mounted 6, 14 or 28 v aircraft lamp; solder lug type terminals; not magnetically shielded; 153 meters are watertight; 150-W also available watertight; ac meters have self-contained rectifier.

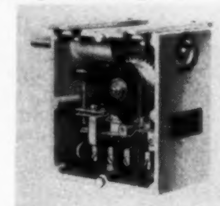
For more data circle MD 9, Page 193

ROTATING LIMIT SWITCH 11

. . . designed for precise adjustment

Furnas Electric Co., 1045 McKee St., Batavia, Ill.

Contact point adjusting screws make precise adjustment of the contact opening time possible.



Designation: L-3464.

Size: 5⅜ in. wide, 4¾ in. deep, 6⅛ in. high.

Service: For limiting travel of rotating parts of machines or equipment by breaking contact at the end of a predetermined number of shaft revolutions (34 max—2 min); 2 normally closed contacts rated 3 amp at 110 v ac, 1.5 amp at 220 v ac, 0.2 amp at 115 v dc with 5 mfd condenser, 0.1-amp at 230 v dc with 0.5-mfd condenser.

Design: Worm gear mounted on a shaft extending through the switch case drives two cams, each of which engages a contact arm to open a circuit at the end of a preset number of shaft revolutions; adjustment is made by changing relative position of cams, one of the cams being spring-loaded and equipped with an indexing-pin which can be placed in any one of 32 holes in the gear body, and by contact adjusting screws which permit precise adjustment of contact opening points.

For more data circle MD 11, Page 193

NYLON-LINED BEARINGS 10

. . . permit dry operation

Thomson Industries Inc., Manhasset, N. Y.

Free-floating nylon liner allows close fits, so that allowance for expansion due to temperature changes is not necessary.

Designation: Thinwall Nylined.

Size and Service: Bearings can operate submerged in many liquids; can also operate dry at high speeds with light loads; 250 F, max. recommended operating temperature.

Bearing	ID (in.)	OD (in.)	Length (in.)	Radial Load Capacity Static (lb)	Radial Load Capacity 3000 rpm (lb)
4N6	¼	11/32	⅜	145	18
5N7	5/16	7/16	7/16	225	21
6N8	⅜	½	½	325	25
8N8	½	⅝	½	435	25
10N12	⅝	¾	¾	900	43
12N12	¾	⅞	¾	1010	43
14N12	⅞	1	¾	1260	43
16N16	1	1⅛	1	2000	52
18N16	1⅛	1¼	1	2260	52
20N16	1¼	1⅜	1	2500	52

Design: Bearing is thin drawn-steel outer sleeve and free-floating Du Pont FM 10001 Nylon liner; ends of sleeve are turned inward to retain liner; compensating gap in liner allows circumferential expansion or contraction with no change in bore diameter.

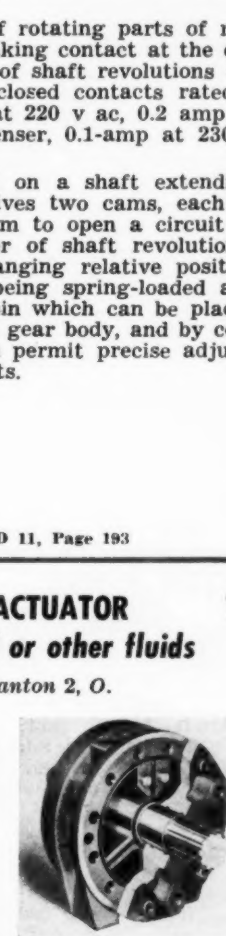
For more data circle MD 10, Page 193

RECIPROCATING ROTARY ACTUATOR 12

. . . operated by air, oil or other fluids

Hydromotor Div., Bonnot Co., Canton 2, O.

Available in wide range of capacities giving equal force in either direction of rotation.



Designation: Hydromotor.

Service: Actuating reciprocating devices; maximum rotation, 100 deg (double-vane type) or 280 deg (single-vane type); operates between −40 F and 260 F:

Model	Rated torque* (lb-in.) at Given Supply Pressure: 100 psi	200 psi	400 psi	600 psi	800 psi	1000 psi
3-2	150	300	600	900	1200	1500
3-4	300	600	1200	1800	2400	3000
3-6	450	900	1800	2700	3600	4500
6-3	1000	2000	4000	6000	8000	10,000
6-6	2000	4000	8000	12,000	16,000	20,000
10-5	5000	10,000	20,000	30,000	40,000	50,000
10-8	7500	15,000	30,000	45,000	60,000	75,000
10-12	11,500	23,000	46,000	69,000	92,000	115,000

*Single-vane type; torque is doubled for double-vane type.

Design: Reciprocating single or double-vane hydraulic motor; single and double-vane shafts are interchangeable in same housing; standard SAE splined shafts, other types available; specials available for temperatures above or below standard; foot or flange mounting.

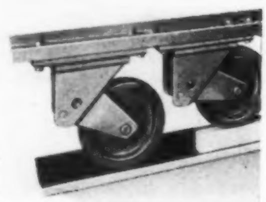
For more data circle MD 12, Page 193

NEW PARTS AND MATERIALS

KNEE-ACTION CASTERS 13

. . . use standard make wheels

All Steel Welded Truck Co., Rockford, Ill.



Knee-action design eliminates shimmy or fishtailing by keeping wheel in constant contact with the floor.

Designation: Clark Duoflex.

Size: 6, 8, 10, 12 in. wheel diam.

Service: Any application where conventional caster is used; capacity, 50 to 1800 lb per caster; minimum wheel deflection is 1/3 of wheel radius; damping or shock absorbing action is proportional to shock and load, keeps wheels in contact with floor at all times.

Design: Arc-welded mild-steel box construction completely encloses knee-action springs; springs have accelerated loading characteristic which is equivalent to having light springs for light loads and heavy springs for heavy loads; lifetime-lubricated oil-impregnated porous bronze bushings; standard wheels of any make may be used.

For more data circle MD 13, Page 193

SEALED TOGGLE SWITCH 15

. . . unaffected by temperature changes

Micro Switch Div., Minneapolis-Honeywell Regulator Co., Freeport, Ill.

Operating characteristics are unaffected by changes in atmospheric pressure or temperature, or corrosive atmosphere, dust, dirt, oil or water.

Designation: 5AT1.

Size: 1 15/16 in. long, 13/16-in. wide, 2 13/16 in. high; mounts in 1/2-in. diam hole; weight, 0.2 lb;

Service:

Load	Voltage (v)	Current (amp)
Motor dc	30	5
Inductive dc	30	10
Resistive dc	30	25
Any ac	125	1

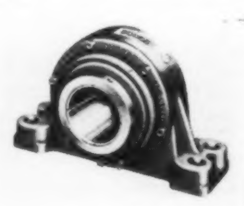
Design: Single-pole, double-throw toggle-actuated switch with normally-open and normally-closed contact; can also be used as normally-open or normally-closed single-throw switch; switch is an assembly consisting of Catalog No. 1HS1 switch designed for aircraft use and toggle actuator.

For more data circle MD 15, Page 193

PILLOW BLOCKS 14

. . . permanently lubricated and sealed

Dodge Mfg. Corp., Mishawaka, Ind.



High radial and thrust load capacities, and stamina to withstand heavy shock loads are features.

Designation: Dodge-Timken All-Steel.

Size:

Shaft diam (in.)	Height (in.)	Width (in.)	Length (in.)
2 5/16, 3	7 1/2	12 1/2	6 1/2
3 3/16, 3 7/16, 3 1/2	9	14 1/2	7 3/4
3 15/16, 4	9 3/4	15 3/4	8
4 7/16, 4 1/2	11 3/8	18	9 1/4
4 15/16, 5	12	19 1/2	9 3/4
5 7/16, 5 1/2	13 1/4	21 1/4	10 1/2
5 15/16, 6	13 3/4	23 1/2	11 1/4
6 7/16, 6 1/2, 6 15/16, 7	15 5/8	26 1/2	12 1/4
7 1/2, 7 15/16, 8	18 3/4	30 1/2	13 1/2
8 15/16, 9	20 3/16	34 1/2	14 3/4
9 15/16, 10	23 1/4	38 1/2	15 1/2

Service: Supporting up to 10 in. diam rotating shafts; self-aligning bearings allow 3 deg shaft misalignment or 5/8-in. total shaft movement at 12 in. distance from center of bearing.

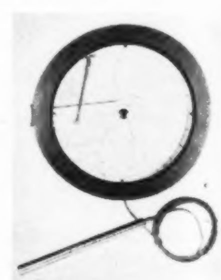
Design: Pillow blocks incorporate self-aligning, double-row Timken tapered roller bearing; bearing may be either expansion or nonexpansion type; double piston-ring seals; shipped fully assembled, adjusted and lubricated.

For more data circle MD 14, Page 193

RECORDING THERMOMETERS 16

. . . record temperatures to —200 F

Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.



Available in a variety of ranges, these recording thermometers provide accurate temperature records.

Designation: NR.

Size: Recorder, 12 in. diam, weighs approximately 10 lb; bulb, 3/4-in. diam, 11 in. long.

Service: Recording temperatures over following standard ranges;

Chart No.	Range (deg F)	Chart Speed (hr)	Chart No.	Range (deg F)	Chart Speed (hr)
2316	−200 to +150	24	2300	−100 to 0	24
2316	−150 to +200	24	2306	−70 to +70	24
2315	−150 to +100	24	2308	−20 to +120	24
2315	−100 to +150	24	2309	−20 to +120	168
2307	−150 to 0	24			

Design: Bourdon-tube type recording thermometer; large brass bulb, which assures maximum heat transfer, is placed in cold chamber; connection between bulb and bourdon tube is 8 ft armored stainless steel tube, longer lengths available; synchronous electric-clock chart drive; special spring drives also available; recording instrument enclosed in aluminum, rubber-gasketed case finished in dark-gray hammered finish; also available as 6 or 8 in. diam indicating thermometer.

For more data circle MD 16, Page 193

NEW PARTS

INDICATOR LIGHT 17

. . . for edge-lit panels

Hetherington Inc., Sharon Hill, Pa.

This light weighs only ¼-oz.

Designation: Type L2000.

Size: 1¼ in. overall length; weight, ¼-oz.

Service: As indicator light for edge-lit AN-P-89 aircraft panels; 6, 12, or 28 v operation with 327 miniature lamp.

Design: Nickel plated brass finished in black to match standard panels; light flange-mounts on backup plate and socket extends through the edge-lit panel; lens, of amber, blue, green, red or white, screws into the socket from front of panel; molded-in terminal will not vibrate or pull loose.

For more data circle MD 17, Page 193

RAINTIGHT SWITCH 19

. . . for outdoor installation

Federal Electric Products Co., 50 *Paris St., Newark* 5, *N. J.*

Front operating handle permits close ganging of these switches.

Designation: Type A.

Service: Outdoor switching; 2 to 5 poles; 30, 60, 100 and 200 amp; 115, 125, 230, 250 v ac; 125, 230, 250 v dc.

Design: Single throw, 2 to 5 pole switch in raintight enclosure; switch mounted on front of enclosure; "stay-up" cover easily secured in horizontal position to facilitate inspection or servicing; arc mufflers on 100 and 200 amp, 250 v switches; box may be padlocked to prevent tampering; fuses are easily removed and inserted; adequate knock-outs are provided; special weatherproof finishes available.

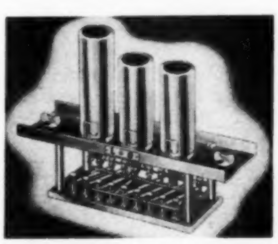
For more data circle MD 19, Page 193

MINIATURIZED PLUG-IN AMPLIFIER 18

. . . weighs only 4.5 ounces

L & O Research and Development Corp., 315 *S.* 15*th St., Philadelphia* 2, *Pa.*

Though light and small, this amplifier delivers approximately 3 w power output.



Service: Driving servomotors where space is at a premium; open loop gain approximately 100,000; delivers approximately 3 w; requires 300 v 35 ma plate supply and 6.3 v at 1 amp for filament; any desired gain can be had by proper selection of feedback resistor.

Design: Plug-in ac feedback amplifier; uses two pentode voltage amplifier tubes and one beam power output tube; models available with either current or voltage feedback; uses JAN components; has Dzus fasteners for quick installation and removal; can be supplied singly or any number in standard chassis with 110 v 400 cycle power supply to meet MIL-C-7452, MIL-P-11268 or AN-E-19 specifications, or with 110 v 60-cycle power supply for bench or industrial use.

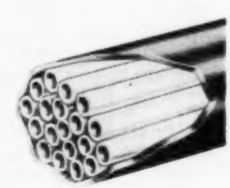
For more data circle MD 18, Page 193

TUBING HARNESS 20

. . . carries bundles of instrument tubing

Samuel Moore & Co., Dekoron Tubing Div., Mantua, O.

Tubing is impervious to attack by moisture or corrosive atmospheres.



Designation: Impervak Poly-Cor.

Size: 4, 7, 10, 14 or 19 tubes, each ¼-in. OD by ⅛-in. ID; $\frac{1}{16}$-in. thick sheath; 500 and 1000 ft coils.

Service: As instrument or control lines; will carry standard instrument line pressures; tubes and sheath are not harmed by acids, alkalies, moisture or corrosive atmospheres.

Design: Bundle of polyethylene tubes over which black polyethylene plastic sheath is extruded; individual tubes are color coded for easy identification; harness is flexible enough to bend around corners, but sufficiently rigid and light to require few supporting brackets; lengths joined with pressure-tight fittings, requiring no tools for assembly.

For more data circle MD 20, Page 193

NEW PARTS AND MATERIALS

MOLDED NYLON GEARS 21

. . . available as stock items

Nylomatic Div., John A. English & Co., Morrisville, Pa.

Lower cost than comparable metal spur gears is claimed.

Size: 1/8-in. face;

Teeth (No.)	Hole Diam (in., stock)	Hole Diam (in., max)*	Teeth (No.)	Hole Diam (in., stock)	Hole Diam (in., max)*
12, 14	3/32	3/32	28, 32	1/8	1/4
16, 18, 20	1/8	1/8	36, 40, 48, 56	3/16	3/8
24	1/8	3/16	60, 64, 72, 80	1/4	3/8

*Not in stock, but can be produced from stock molds on short notice.

Service: For light loads; operate quietly; allow relatively large tolerance in center location.

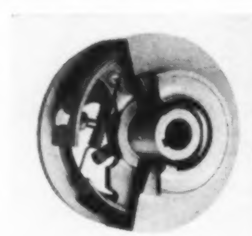
Design: Injection-molded nylon spur type, 48 diametral pitch, 14½ degree pressure angle; gear-pinion combinations also available.

For more data circle MD 21, Page 193

CENTRIFUGAL CLUTCH 22

. . . provides adjustable, automatic cut-in

Farm-Easy Products Inc., 1208 *E. Centennial Ave., Muncie, Ind.*



Having less than a dozen parts, this clutch is adjusted by moving clutch-band tension spring.

Designation, Size and Service: For automatic load cut-in at predetermined speed; provides "load-free" starting of gasoline engines; lightweight, minimizing shaft deflection; releases automatically before stall speed;

Specification	Model 4500	Model 5000	Model 8000
Power (hp)	½–3	2½–6	5–12
Shaft Diam. (in.)	5/8, 3/4	5/8, 3/4, 1, 1⅛	3/4, 1, 1⅛, 1¼
Pulley Size*	3, 3½, 4	3, 3½, 3	3, 3½, 4, 5, 7
Max Torque (lb-ft)	5	15	23
Cut-in Speed (rpm)	1000–1500	1000–1200	600–1200

*Single groove (multiple pulleys available).

Design: Automatic centrifugal; at cut-in speed, centrifugal force expands clutch band to engage drum; cut-in speed is adjustable by adjusting band tension spring, which normally holds clutch out of engagement; attaches with 1 or 2 set-screws and keyway to shaft of any standard 4-cycle gasoline engine; AB-section V-belts; Oilite bearings, or standard ball or roller bearings, special.

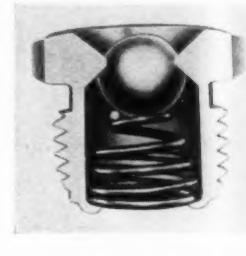
Application: Portable power saws and mowers; conveyors; cement mixers; winches; motor scooters; garden tractors; pumps.

For more data circle MD 22, Page 193

GREASE FITTINGS 23

. . . flush-type with ball check

Universal Lubricating Systems Inc., 672 *Allegheny Ave., Oakmont, Pa.*



Possibility of knocking off this fitting with a chance blow or collision is minimized.

Designation and Size:

Part No.	Length (in.)	Thread (in., pipe)	Part No.	Length (in.)	Thread (in., pipe)
915	11/32	3/8	2910†	9/16	1/4
2900	13/32	3/16*	2920†	11/32	1/8

*Not threaded—diam for force fit. †Screw-driver slot.

Service: Grease lubrication without oil or grease leakage due to back pressure; free flow, since passage above ball at point of fullest depression is less than passage through bottom of fitting; offers little obstruction or opportunity for collision and minimum chance of being damaged or knocked off.

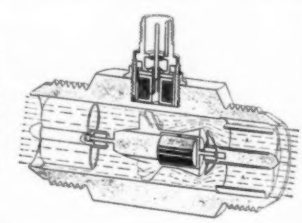
Design: Flush type ball check.

For more data circle MD 23, Page 193

FLOWMETER 24

. . . for accurate flow measurements

Minneapolis-Honeywell Regulator Co., Industrial Division, Wayne and Windrim Aves., Philadelphia 44, Pa.



Accurate measurement and recording of flow of liquids or gases at temperatures of −269 C to +1200 F, and pressures up to 20,000 psi, are possible.

Designation: Potter-Brown.

Size: Sensing elements, 3/16 to 6 in. dia.

Service: Measuring and recording flow of liquids and gases, including toxic, highly corrosive and viscous liquids, at temperatures of −269 C to +1200 F and pressures to 20,000 psi; measures flow from 0.09 to 4000 gpm; sensing element can be mounted horizontally or vertically.

Design: Sensing elements, designed for installation in flow line, have pickup coil mounted on the outside, and rotor with internally mounted permanent magnet; rotor spins at a rate proportional to flow velocity, inducing an alternating voltage in the pick-up coil; venturi design of rotor causes it to float between upstream and downstream guide bushings, reducing friction; voltage induced in pickup coil is conducted to electronic converter which converts the alternating voltage to direct voltage; direct voltage is fed to potentiometer for measuring and recording.

For more data circle MD 24, Page 193

NEW PARTS AND MATERIALS

ZINC COATING 25

... may be dipped, sprayed or brushed

Industrial Metal Protectives Inc., 401 *Homestead Ave., Dayton, O.*

Withstands atmospheric, subsoil and underwater corrosion, and abrasion.

Designation: Zincilate 100 and 300.

Form: Liquid vehicle and powdered pigment.

Service: Protecting iron and steel from corrosion; coating is unaffected by gasoline, kerosene or other petroleum products; withstands hot water to 150 F and air temperatures to 600 F; unaffected by mild acids or alkalies; color coats may be applied as final finish; 100 covers 400 to 550 sq ft per gal, No. 300 covers 600 to 800 sq ft per gal.

Properties: Neutral gray-colored liquid, when mixed, is applied like paint; 100 must be baked, 300 is for air drying.

For more data circle MD 25, Page 193

HERMETICALLY SEALED RELAY 26

... is not damaged by 125g shock

Electrical Products Corp., Relay Div., 1100 *N. Main St., Los Angeles* 12, *Calif.*

Weighing 4.8 oz, this relay is claimed to be one of the lightest available for its rating.

Designation: A52-111.

Size: $2\frac{1}{16}$ in. by $1\frac{13}{16}$ in. by $1\frac{9}{16}$ in.; $2\frac{3}{8}$ in. between mounting hole centers; weight, 4.8 oz.

Service: Rated 10 amp at 28 v dc or 115 v 400-cycle ac; shock, vibration and acceleration resistant; free of resonance over a range of 5 to 200 cycles per sec in vibration tests; withstands 125g shock.

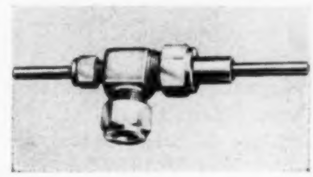
Design: Double-pole, double-throw relay using rotary armature which has been AN approved in larger enclosures; design to meet MIL-R-6106 specification; hermetically sealed.

For more data circle MD 26, Page 193

HEAT EXCHANGER TEE 27

... for jacketing of fluid lines

Crawford Fitting Co., 884 *E.* 140*th St., Cleveland* 10, *O.*



Available in a variety of materials and sizes, this tee has numerous applications in laboratories and process industries.

Designation: Swagelock.

Size: Process line tube—1/8 to 3/4-in. OD; jacket tube —3/8 to 1 in. OD.

Service: Connecting concentric jacketing tubes for cooling or heating; usable over a wide range of temperatures dependent upon material of tee and heat exchange media.

Design: Tubing tee which is bored to allow small tubing to pass completely through the jacketed end; available in brass, aluminum, steel, stainless steel and monel; uses double-ferrule principle, no flaring or other preparation of tubing necessary.

For more data circle MD 27, Page 193

SYNCHRONOUS INVERTER 28

... operates from 0 to 3500 cycles

The Bristol Co., Waterbury 20, *Conn.*

Armatures of such low mass that their mechanical response is practically instantaneous allow this inverter to operate up to 3500 cycles.

Designation: Syncroverter Switch.

Size: $2\frac{3}{4}$ in. high, $2\frac{5}{8}$ in. diam.

Service: Converting dc signals as small as 0.05 mu v to ac at frequencies as high as 3500 cycles per sec; ac then can be transformed or amplified and applied to electronic, electrical or servo systems; dissymmetry is less than 0.5%; vibration and shockproof.

Design: Synchronous inverter in hermetically sealed can; low-mass armatures; can also be furnished with two independent sets of single-pole double-throw contacts.

For more data circle MD 28, Page 193

THERMOSETTING PLASTIC 29

. . . will not bleed when immersed in water

Plaskon Div., Libbey-Owens-Ford Glass Co., Toledo 6, O.

Low cost is one of the features claimed.

Designation: Black 107, Brown 120.

Form: Granulated molding material.

Service: Molded plastic has high dielectric strength and arc resistance; highly resistant to grease, oil and solvents; withstands temperatures to 170 F.

Properties: Thermosetting urea plastic; tasteless, odorless and inert; no pigment bleeding in water; low rate of water absorption; light-fast;

Dielectric strength (v/mil)		Arc resistance (sec)..	85-95
short time, ⅛-in.	300-400	Loss factor	
step by step, ⅛-in.	250-300	at 60 cycles.......	0.28-0.45
Dielectric constant		at 10^6 cycles	0.19-0.28
at 60 cycles ...	7.0-9.5	Compression strength (psi)	25,000-35,000
at 10^6 cycles ..	6.4-7.0	Tensile strength (psi)	5000-10,000
Dissipation factor		Flexural strength (psi)	10,000-16,000
at 60 cycles....	0.040-0.050	Impact strength, Izod (ft-lb per in. notch)	0.25-0.35
at 10^6 cycles ..	0.030-0.040		

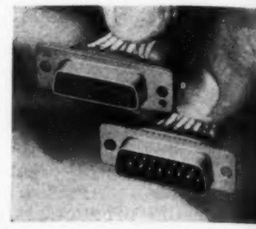
Applications: Lighting fixtures; food machinery; refrigerator hardware; electrical devices; electrical components and housings.

For more data circle MD 29, Page 193

MINIATURE CONNECTORS 31

. . . use nylon insulation

Cannon Electric Co., 3209 Humboldt St., Los Angeles 31, Calif.



Suitable for rack and panel mounting, these connectors may also be used as plugs.

Designation: Series D.

Size: DA-15 (15 contacts), 1 13/32 long, 43/64 wide, weight 0.027 lb; DB-25 (25 contacts), 2 5/64 long, 43/64-in. wide, weight, 0.054-lb; DC-37 (37 contacts), 2 23/32 in. long, 43/64-in. wide, weight, 0.070-lb; DD-50 (50 contacts), 2⅝ in. long, 59/64-in. wide, weight, 0.075-lb.

Service: For making as many as 50 electrical connections; takes No. 20 AWG, B & S stranded wire; rated at 5 amp per contact; flashover voltage, 1414 v dc, 1000 v 60-cycle ac (rms).

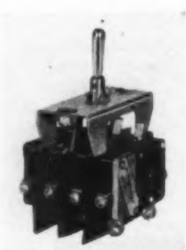
Design: Subminiature connector; steel shell has strength comparable to usual aluminum shell and lighter weight; FM-10001 Nylon insulator with high dielectric strength makes the use of many contacts in small space possible; keystone shape of connectors prevents mismating of connectors; junction shells available, permit use as movable cord plugs; contacts are copper-base alloy, gold plated.

For more data circle MD 31, Page 193

TOGGLE SWITCH ASSEMBLY 30

. . . 12 switches may be operated by one lever

Micro Switch Div. of Minneapolis-Honeywell Regulator Co., Freeport, Ill.



Twelve switches complete with actuator weigh only 6.5 oz.

Designation and Size: Length, 2¼ in., mounting hole diam, 15/32-in. for all assemblies;

No.	No. of Switches	Lever Positions*	Width (in.)	Height (in.)	Weight (oz)
4AT4	4	M, M	1 1/16	3 1/16	4.0
4AT5	4	NM, NM	1 1/16	3 1/16	4.0
4AT6	4	M, NM	1 1/16	3 1/16	4.0
7AT4†	6	M, M	1½	3 1/16	4.5
8AT1†	8	NM, NM	1 53/64	3 7/16	5.0
10AT1†	10	NM, NM	2 13/64	3 7/16	6.0
Special†	12	NM, NM	2 13/16	3 7/16	6.5

*M—Maintained; NM—Not maintained. †Available as specials in all other lever positions.

Service: Individual switch ratings—10 amp at 30 v dc resistive, 10 amp at 30 v dc inductive at sea level; 6 amp at 30 v dc inductive at 50,000 ft; 10 amp at 125 or 250 v ac inductive or resistive.

Design: Micro switches with toggle actuator; basic switch is small snap-action SPDT with beryllium copper spring, silver contacts and heavy-gage brass screw terminals; median life of 52,000 operations tents in both extreme positions also have center detent; if only one or neither extreme lever position is detented, lever is self-returning to neutral.

For more data circle MD 30, Page 193

MINIATURIZED DC RELAY 32

. . . weighs only three ounces

Radio Corp. of America, RCA Victor Div., Tube Dept., 115 S. Fifth St., Harrison, N. J.

Inert-gas filled and hermetically sealed, this relay is said to give long and reliable service though subjected to extremes of temperature, humidity, shock, and vibration.

Designation: RCA-203W1.

Size: 1 43/64 in. diam plus 45/64-in. for mounting lugs; 1¾ in. high; 1 9/16 in. center to center of mounting studs; weight, 3 oz.

Service: Withstands 50*g* acceleration for a period of 10 millisec, vibration of 0.06-in. amplitude and 10 to 55 cycles per sec for a period of 2 hr; life of 100,000 operating cycles at 12 cycles per min with rated load;

Ambient temperature range	—55 to +85 C
Normal dc operating voltage	26.5 v
Maximum dc pull-in voltage	18.0 v
Maximum dc drop-out voltage	13.0 v
Resistive load per contact at 26.5 v	2 amp
Inductive load per contact at 26.5 v	1 amp
Dc coil resistance at 25 C	230 + 10% ohm
Dc contact resistance, max	0.070 ohm

Design: Six-pole, double-throw dc relay designed to meet MIL-R-5757 specification; has palladium contacts in break-before-make arrangement; can be operated in any position; hermetically sealed to prevent entry of dust and moisture, and to prevent corrosion.

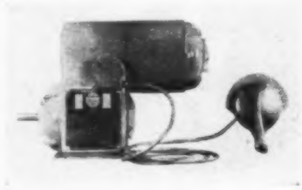
For more data circle MD 32, Page 193


DON'T
COMPROMISE
FOR JUST
A BEARING
be sure
it is THE
bearing!
"THE BEARING" will probably be a Johnson Sleeve Bearing, designed specifically for the application. There are several important factors involved in the selection of the correct bearing for any job . . . principally service conditions. Operating speed, temperatures, corrosive conditions, load, shock . . . all must be carefully considered. Since Johnson Bronze produces all types of sleeve bearings, uses all types of bearing metals, their engineers will give you unbiased advice and will assist you in designing the correct sleeve bearing for your application. Write, wire or phone for an appointment with a Johnson Bearing Specialist.
JOHNSON BRONZE COMPANY
525 South Mill St., New Castle, Pa.
BRONZE-ON-STEEL
—copper lead
STEEL BACK
—babbitt lined
BRONZE BACK
—babbitt lined
CAST BRONZE
ALUMINUM ALLOY
LEDALOYL
—powder metallurgy
JOHNSON JB BEARINGS
Sleeve-Type
SLEEVE BEARING HEADQUARTERS SINCE 1901

NEW PARTS AND MATERIALS

VARIABLE SPEED MOTOR 33
. . . has flexible shaft control

U. S. Electrical Motors Inc., Box 2058, Los Angeles 54, Calif.



Control handwheel indicates motor speed, and may be located at a distance from the motor.

Designation: 5 VA.

Size: ¼, ⅓, ½, and ¾ hp.

Service: Speed changing; speed ranges, 4 to 10,000 rpm, ratio up to 10:1.

Design: Variable speed electric motor incorporating variable pitch pulley and belt drive; drive controlled by flexible shaft which allows control handwheel to be located 5 ft or more from motor.

For more data circle MD 33, Page 193

PHOSPHATE FINISH 35
. . . cleans and coats aluminum and steel

Rossaul Co., 170 Fifth Ave., New York 10, N. Y.

Removes light rust, tarnish and light oil, and leaves a phosphate coating.

Designation: 57.

Form: Liquid.

Size: 13-gal carboys; 1 gal units.

Service: Light cleaning of aluminum and steel, and deposition of white-gray phosphate coating as base for subsequent primer or color coat; alumina oxide surface, which protects aluminum, is not penetrated; applied by wipe-on or still-tank dip; still-tank dip requires precleaning of metal.

Properties: Phosphoric-base; diluted to 25% solution for wipe-on application, 10% for dip-tank.

For more data circle MD 35, Page 193

PIN FASTENER 34
. . . now available in larger sizes

Self Lock Fastener Corp., 259 Stephens St., Belleville, N. J.

Can be used in place of tapered pins in tapered holes for lock fastening.

Designation: Sel-Lock.

Size: Standard and light-duty in 0.312, 0.375 and 0.500-in. diam.

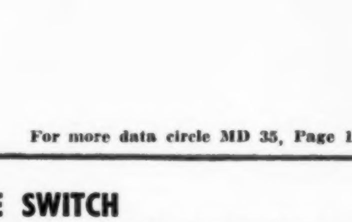
Service: Lock fastening under shock and vibration conditions; standard-duty has shear strength equal to solid pin of same diam; light-duty is for fastening unlike materials; can replace tapered pins; require only drilled straight hole; may be inserted by hand or automatic machines.

Design: Slotted dowel pin, of carbon or type 420 stainless steel; blue-black oiled finish on carbon steel, bright gray on type 420 stainless steel; hardness—carbon steel, Rockwell 46-53C—stainless, Rockwell 43-52C; chamfered on both ends for easy insertion; special materials—copper alloys, aluminum alloys, nonheat-treatable stainless steels; special finishes—cadmium or zinc plate, with or without postplate treatments; also available in smaller sizes.

For more data circle MD 34, Page 193

TOGGLE SWITCH 36
. . . cylindrical design reduces size

Hetherington Inc., Sharon Hill, Pa.



Twelve of these switches can be mounted in the space required for 9 conventional rectangular switches.

Designation: T 1000.

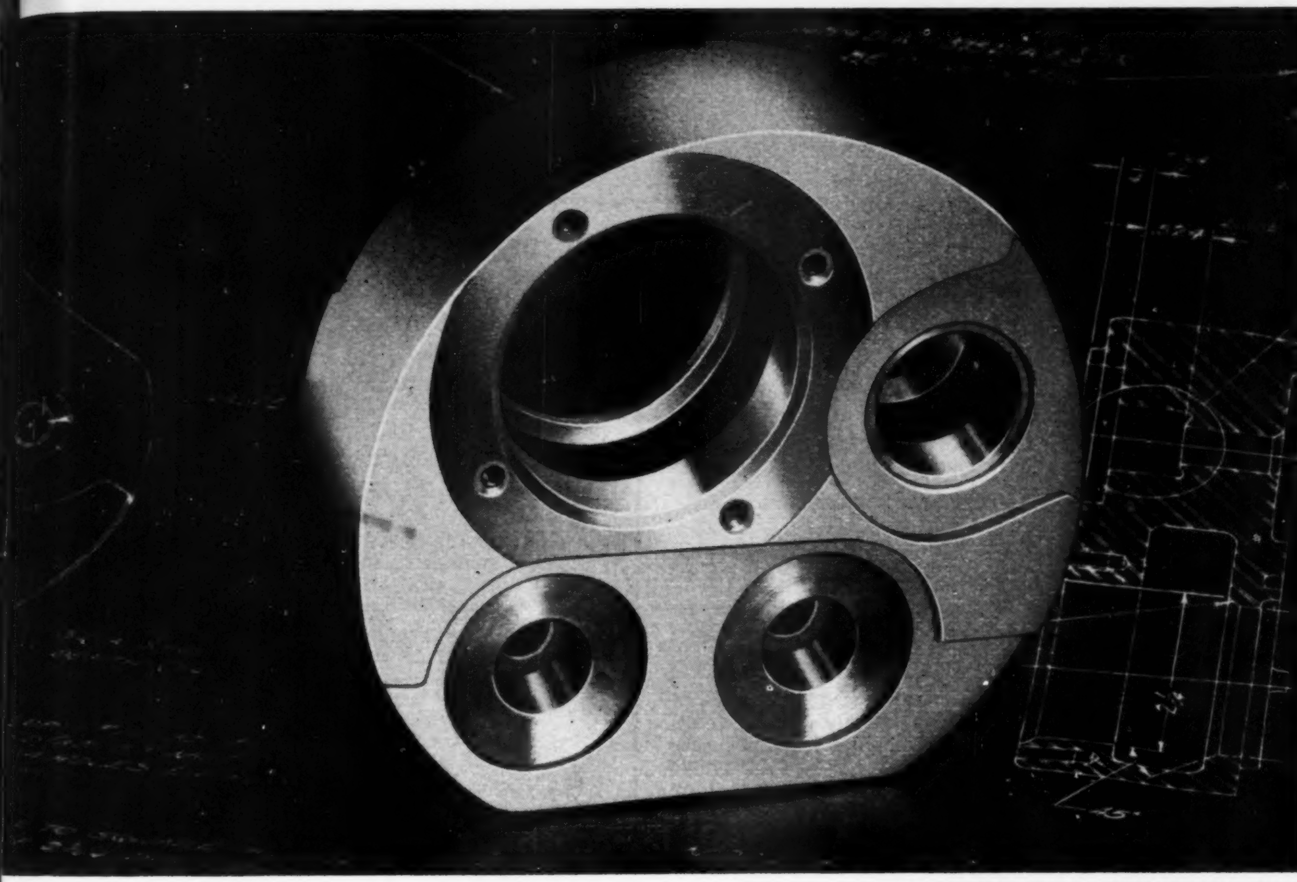
Size: 2$\frac{3}{32}$ in. long, ¾-in. diam; weighs 1 oz; mounts in ½-in. diam hole.

Service: Rated 10 amp at 115 v ac, 20 amp at 28 v dc (resistive load), or 15 amp at 28 v dc (inductive load); operates at 50,000 ft altitude; will withstand 50g shock; temperature range, −65 to +180 F.

Design: Single-pole double-throw switch, also available as momentarily on, momentarily off, or on-momentarily on type; cam-roller type contactor assures snap action and gives effective contact wiping; contacts are heavily silvered copper; insulating spacer prevents arcing between terminals; No. 6 screw terminals used.

For more data circle MD 36, Page 193

Only MERCASTING* can make this as a unit!



*FROZEN MERCURY INVESTMENT CASTING

PROBLEM:

Three expensive machined parts were formerly required to get the necessary internal cavities for this valve body. Use on an electronic tube evacuator made special tolerances necessary. After assembly by welding, re-machining, and testing, a large part of the total production was rejected for failure to meet the rigid specifications of the manufacturer—Distillation Products Industries, Division of Eastman Kodak Company, well-known producer of high-vacuum equipment.

SOLUTION:

DPI brought the problem to Alloy Precision Castings Company. Cooperating Alloy foundry engineers and metallurgists developed the pictured one-piece, 5½-lb. mercasting of 303 stainless. The "as-cast" piece can be held to an average plus or minus .003 inch per inch on certain critical dimensions. A comparatively small amount of finish machining is needed on valve seats and tapped holes.

A-6298

SUMMARY:

The illustrated mercasting is a better valve body easily produced in a more desirable material. Tremendous savings are effected through elimination or drastic reduction of special tooling, machining, waste material, and scrapped production.

Send prints of your problem parts today. Learn how Alloy Precision can save you time and money.

ALLOY PRECISION CASTINGS COMPANY, DEPT. B-3
45th and Hamilton, Cleveland 14, Ohio

Please RUSH Bulletin 706 describing the Mercasting Process.

NAME ____________________

TITLE ____________________

COMPANY ____________________

ADDRESS ____________________

CITY ____________________

ZONE ________ STATE ____________________

ALLOY PRECISION *Castings Company*

EAST 45th ST AND HAMILTON AVE. • CLEVELAND 14, OHIO

ENGINEERING DEPARTMENT

EQUIPMENT

For additional information on this new equipment, see Page 193

LEAD HOLDER 37

. . . available with lead pointer

Alvin Co., Dept. R, Windsor, Conn.

Leads are easily sharpened by a few turns in pointer.

Size: Lead holder, approximately 5 in. long; lead pointer, 2¾ in. high, 1½ in. diameter.

Service: For general purpose drafting; holder takes any standard size drawing lead; push-button lead ejection to desired length; pointer can be used with most standard lead holders, retains lead dust; pointer cutting blades are changeable.

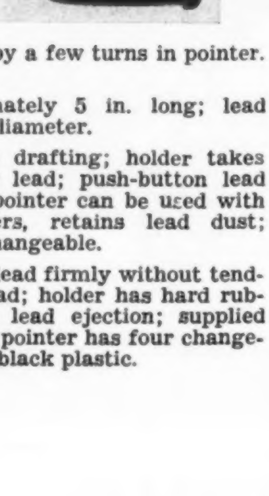
Design: Collet in holder grips lead firmly without tendency to shear or weaken lead; holder has hard rubber barrel, coil spring for lead ejection; supplied with or without pocket clip; pointer has four changeable knife blades, is made of black plastic.

For more data circle MD 37, Page 193

DRAFTING TABLES 38

. . . in three all-steel models

Stacor Equipment Co., 768-778 *E. New York Ave., Brooklyn* 3, *N. Y.*



Four-post design with steel footrest provides firm supporting base for these tables.

Size: Drawing surface, 36 x 60 in., 36 x 72 in., 42 x 72 in; shallow reference drawer, 36 x 27½ x 2 in.; tool drawer, 9¾ x 27 x 5 in.

Service: Top is adjustable by use of 2 steel adjusting devices; shipped knocked-down.

Design: Smooth-finish kiln-dried soft-wood top; three models, (1) with shallow reference drawer and locking tool drawer, (2) with tool drawer only, (3) without drawers; Prestwood dust cover; steel frame, adjusting devices, and drawers; all models available in the different board sizes; hard-baked grey-enamel finish.

For more data circle MD 38, Page 193

PHOTOGRAPHIC PAPER 39

. . . produces positives from positives

Eastman Kodak Co., Rochester 4, *N. Y.*

Reduced or enlarged positives may be made directly from drawings or records with this paper.

Designation: Kodagraph Projection Positive.

Size: Rolls, 30, 36, 42 in. wide by 100 ft long; sheets, 8½ by 11 in.

Service: For making positive paper prints; can be used in process camera for direct reduction positives from large drawings, in enlarger to produce positive prints from positive microfilm records.

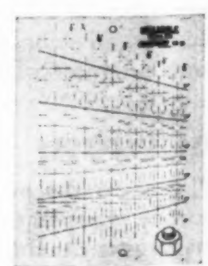
Design: Prints are as legible as large drawings; requires special developer.

For more data circle MD 39, Page 193

PLASTIC TEMPLATE 40

. . . for hex-nuts and hex-head screws

Rapidesign Inc., P. O. Box 592, *Glendale, Calif.*



Cutouts are provided for drawing chamfercircles and flats, as well as hexagons, in six elevations.

Designation: 121.

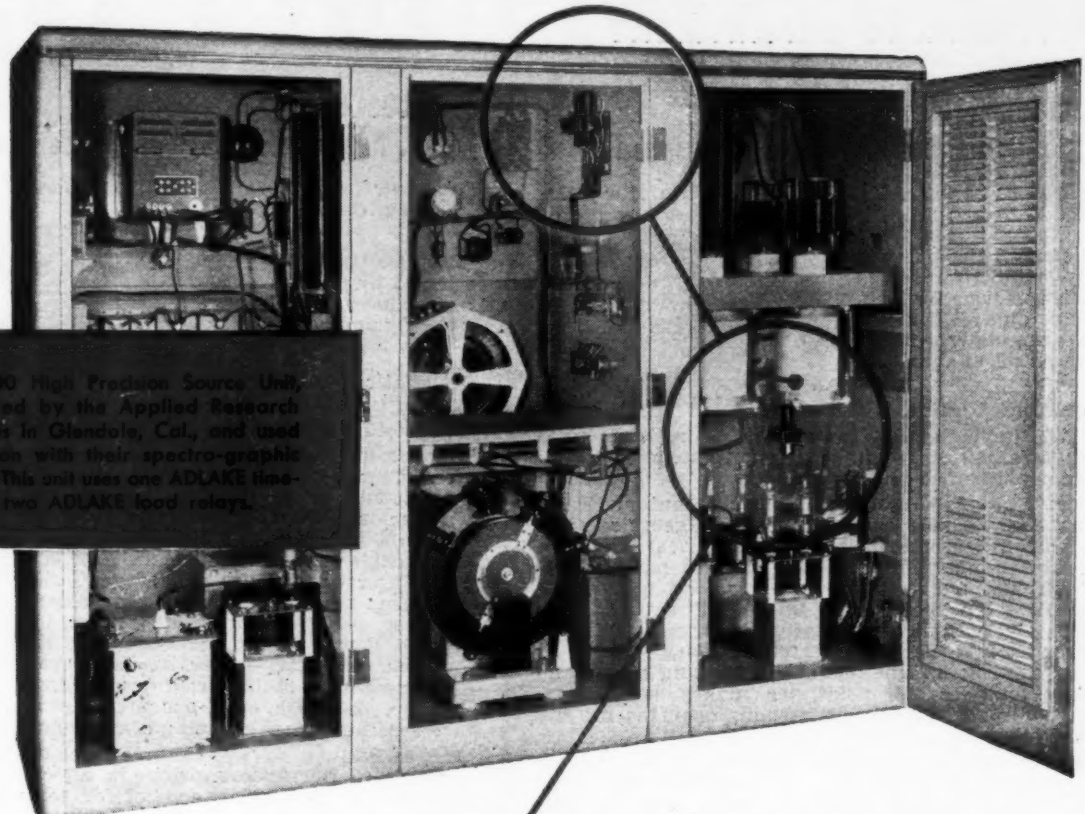
Size: 8½ in. long, 5¾ in. wide, 0.030-in. thick.

Service: For general-purpose drafting or technical illustrating; covers 6 screw sizes from No. 10 to ½ in.; $\frac{1}{16}$-in. increments; 6 elevations—90, 60, 45, 30, 15 and 0 deg; pencil allowance for accuracy.

Design: 103 cutouts arranged so that hexes, chamfer circles and flats can be drawn for each size and elevation; cutouts are precision smooth; matte finish plastic.

For more data circle MD 40, Page 193

ADLAKE RELAYS AT WORK—One of a series of advertisements on specific ADLAKE installations.



Model 4700 High Precision Source Unit, manufactured by the Applied Research Laboratories in Glendale, Cal., and used in connection with their spectro-graphic equipment. This unit uses one ADLAKE time-delay and two ADLAKE load relays.

for assured sensitivity in spectro-chemical analysis

HIGH PRECISION SOURCE UNITS USE Adlake MERCURY RELAYS

EVERY ADLAKE RELAY GIVES YOU THESE ADVANTAGES:

- **HERMETICALLY SEALED**—Dust, dirt, moisture, oxidation and temperature changes can't interfere with operation
- **SILENT AND CHATTERLESS**
- **REQUIRES NO MAINTENANCE**
- **ABSOLUTELY SAFE**
- **MERCURY-TO-MERCURY CONTACT**—Prevents burning, pitting and sticking

Adlake TRADE MARK

95th year of serving the transportation and building industries

Sensitivity is a feature of ARL's High Precision Source Unit, which provides very high intensity discharges for the direct reading of the entire range of concentrations, with a high degree of reproducibility.

And sensitivity, of course, is one thing for which ADLAKE Relays are famous. Sensitivity, coupled with complete dependability and maintenance-free operation, makes them a natural for every exacting application.

Whatever your relay problem may be, ADLAKE can help you solve it—either with one of the models in the wide ADLAKE line, or with a special design worked out specifically for you. Write The Adams & Westlake Company, 1128 N. Michigan Ave., Elkhart, Indiana, for full information.

THE Adams & Westlake COMPANY

Established 1857 • ELKHART, INDIANA • New York, Chicago

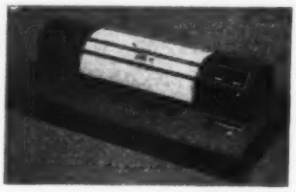
Manufacturers of ADLAKE Hermetically Sealed Relays

ENGINEERING DEPARTMENT **EQUIPMENT**

GRAPH PLOTTER 41

. . . records digital data at high speed

Logistics Research Co., 141 South Pacific Ave., Redondo Beach, Calif.



Instrument is particularly suitable for use as read-out device for electronic digital computers, especially digital differential analyzers.

Designation: Logrinc.

Size: 7¾ in. high, 12½ in. wide, 25⅛ in. long; plotting area, 12 x 18 in., continuous strip of 12 in. wide paper can also be used.

Service: For plotting digital outputs; automatically plots one variable against another algebraically in incremental steps in response to electrical impulses; operates directly from sensing devices having digital output; bidirectional movement on both axes; plots 20 steps per sec in all four directions; step size, $\frac{1}{32}$-in. with no accumulative error; self-contained power supply; electronic, external, remote switch or relay control; instant manual positioning of pen and drum; long-life ball point pen makes several carbon copies or duplicating stencil; operates on any plane; power source, 110 v ac.

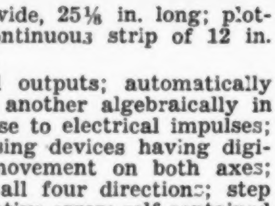
Design: Independent action of x and y axes; input on 4 terminals, $+x$, $+y$, $-x$, $-y$; preferred input from contacts of external switch or relay; stepping motors drive axes through ratchets and pinion gears; motor couplings spring loaded to prevent backlash; removable motor units require no lubrication; spring-loaded pen cartridges available in different colors; automatic interlock prevents carriage overtravel.

For more data circle MD 41, Page 193

PRESSURE TRANSDUCER 42

. . . has 1,000,000: 1 pressure range

Beta Corp., P.O. Box 8625, Richmond 26, Va.



Sensitivity of this device extends to sound pressures as low as 0.0001-psi.

Designation: EK-3.

Size: Overall length, 1¾ in.; barrel, 1⅝ in. diam.; flange, 2¼ in. diam by ¼ in. thick.

Service: For measuring fluctuating or transient pressures over extended ranges of amplitude and frequency; frequency response with 2-ft cable; flat within 3 db from 0.25-cps to 30 kc; pressure range, 0.0001 to 100 psi; sensitivity, 250 mv per psi minimum; self-generating; will withstand up to 10 psi steady pressure without damage; operating pressure limits, 10 × frequency in cps for alternating pressures, 1/duration in seconds for pulse pressures; temperature range, −20 to +60 C ambient; sensitivity at midband frequencies unaffected by cable length.

Design: Operation principle makes use of streaming potential developed by polar liquid flowing through porous plug; output impedance, 100,000 ohms max; sensitivity calibration, ±2% at 25 C; linearity, within 1% over any 10 to 1 pressure range; recessed diaphragm protected by wire mesh; mounting, 3 flat head screws on 2 in. diameter circle; available in different shapes with auxiliary equipment.

For more data circle MD 42, Page 193

OSCILLOSCOPE 43

. . . has seven-inch viewing screen

Teletronic Laboratories, 1835 W. Rosecrans Ave., Gardena, Calif.

Picture tube is protected by metal shield, reproduces waveshapes clearly and accurately.

Designation: 101.

Size: 10½ in. high, 19 in. wide, 15½ in. deep; weight, approx 55 lb.

Service: For general-purpose laboratory work; vertical amplifier has 5 stage push-pull amplification, sensitivity of 10 mv peak-to-peak per in.; vertical sinusoidal frequency response, 0-200 kc within 10% down, 0-500 kc within 6 db down from max response; vertical expansion to 5 times full scale; horizontal amplifier has 3 push-pull stages, sensitivity of 100 mv rms per in.; horizontal sinusoidal frequency response, 0-100 kc within 10% down, 0-300 kc within 6 db down; horizontal expansion to 6 times full screen diameter; linear sawtooth voltage provided by time-base oscillator; retrace elimination; sweep range, 2-30,000 cps; synchronizer assures steady sweep; functional grouping of controls.

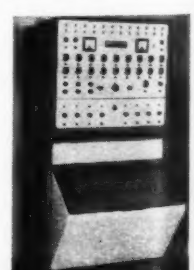
Design: Built in voltage calibrator for dc and ac readings; 1% tolerance resistors in all critical circuits; horizontal positioning control for selective viewing; steel cabinet finished in black wrinkle; black-finished panel with baked-on labels; bright chrome chassis; provision for standard rack mounting.

For more data circle MD 43, Page 193

DIFFERENTIAL ANALYZER 44

. . . solves nonlinear differential equations

Goodyear Aircraft Corp., Dept. 65A, Akron 15, O.



One unit can solve 12th-order differential equations involving 10 initial conditions.

Designation: L3 GEDA.

Size: Approximately 72 x 34 x 30 in.; weight, 875 lb.

Service: For solving ordinary nonlinear differential equations or for studying boundary value problems and devices having static or coloumb friction, backlash, velocity, acceleration limiting, etc.; problem solution may be stopped and held when variable reaches preset level, when 2 variables become equal, or when computer element fails; automatic amplifier stabilization; automatic error indication; all connections through removable problem board; units can be paralleled for complex problems; operation from front in standing, sitting, remote positions; 12 initial problem conditions; up to 165 independent parameter settings; power source, 105-125 v, 60 cycle, single phase.

Design: 24 dc universal-computing amplifiers; 12 integrating capacitors; low grid current and noise; wide amplifier band width; front panel space for built-in auxiliary equipment; built-in precision voltmeter; easy access to all units for maintenance; reinforced plastic problem board; cooling system; up to 36 scale-factor potentiometers; mobile.

For more data circle MD 44, Page 193

simple, easy to use!

speed detailing and specifying!

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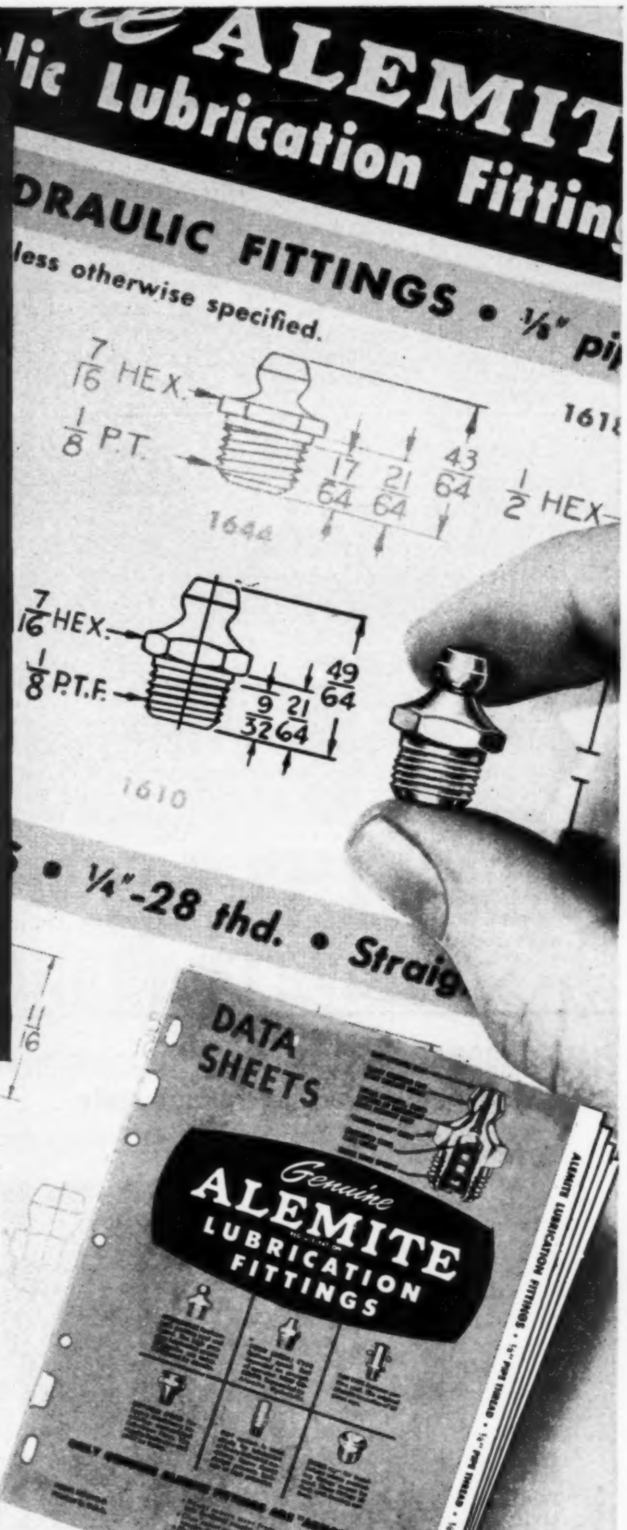
ALEMITE LUBRICATION FITTING DATA SHEETS



They're here! Available to designers and draftsmen for the first time! A handy, *complete* grouping of the most *complete* line of fittings on the market. Genuine Alemite Lubrication Fittings . . . highest quality fittings from the leader and pioneer in the field of lubrication!

They're here! *Fully detailed. Easy-to-use.* With every fitting illustrated actual size—complete with individual dimensions—ready for transfer to your own blueprints. *No complicated cross reference charts or procedures are involved.*

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Please rush me FREE your brand new, up-to-the-minute Alemite Lubrication Fitting Data Sheets.

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ENGINEERING DEPARTMENT EQUIPMENT

PYROMETER 45

. . . combined with dc millivoltmeter

Thermo Electric Mfg. Co., 480 *Huff St., Dubuque, Iowa*

This portable instrument has three scales calibrated in Fahrenheit, Centigrade and millivolts.

Designation: 17, 35, 50.

Size: Case, 7 in. wide, 9 in. long, 4½ in. high; weight, 3 lb; thermocouple, 18 in. long.

Service: For general purpose use in measuring temperatures or dc mv; thermocouple and leads permit readings up to 8 ft from instrument; accuracy within 1% of full scale deflection; approximate scale ranges for 3 models;

Model	Temperature (F)	Temperature (C)	Voltage (mv)
Model 17	32-800	0-430	0-17.5
Model 35	32-1600	0-870	0-36
Model 50	32-2250	0-1230	0-50

Design: D'Arsonval type movement; Alnico V magnet; polished pivots; jewel bearings; automatic compensation for internal resistance changes due to ambient temperature variations; fluid thermometer beside meter indicates proper cold junction setting; thermocouple has welded tip, joins to lead wires in metal housing; thermocouple and lead wires of chrome-alumel; sturdy carrying case with detachable cover.

For more data circle MD 45, Page 193

STOPCLOCK 47

. . . with three-way timing control

Andrew Technical Service, 3805 *N. Clark St., Chicago* 13, *Ill.*

Rugged and durable, this low-cost instrument can be used for repetitive or accumulative timing.

Designation: Durachron.

Size: Dial, 4 in. diam.

Service: Three-way timing control, (1) stop and go timing without return to zero, (2) conventional repetitive-timing in which hand is stopped and returned to zero for new start, (3) snapback timing in which hand is returned to zero but motion is continuous without dead stop; stopwatch precision; large dial for easy reading; frees hands during timing operation; sweep hand has 60-sec travel, recording hand has 60-min travel.

Design: Spring-wound for portability; control lever at side stops and starts sweep hand and recording hand; stem at top is pressed to return both hands to zero; glass dial front; heavy metal case.

For more data circle MD 47, Page 193

OSCILLOGRAPH 46

. . . has direct reading voltage scale

Allen B. Du Mont Laboratories Inc., Instruments Div., 1500 *Main Ave., Clifton, N. J.*

Built-in calibration system permits measurement of any portion of input-signal amplitude.

Designation: 304-A.

Size: 13½ in. high, 8¾ in. wide, 19½ in. deep; weight, 50 lb; 5 in. screen.

Service: For general purpose qualitative or quantitative oscillograph analysis; operation parallels that of vacuum-tube voltmeter; minimum scan, ±2 in. from screen center on both axes; maximum scan, 30 in. horizontal, 20 in. vertical; sensitivity, 25 peak-to-peak mv per in. maximum on *y* amplifier, 0.3 peak-to-peak v per in. on *x* amplifier; measures 0-1000 v directly on illuminated, calibrated screen scale; astigmatism control; frequency response, less than 10% down at 100 kc or 50% down at 300 kc for both amplifiers; recurrent and driven sweeps of variable frequency 2-30,000 cps; extra-low-frequency sweeps possible; sync limiting on sweeps; power, 115-230 v, 110 w, 50-400 cycles.

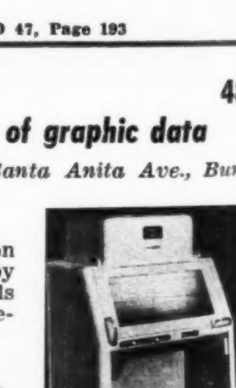
Design: Built-in calibration system; precision attenuator with resistors ±1% accurate; flat-face cathode-ray tube; ac or dc amplification; *y* amplifier input heaters are regulated; calibration lines, 5 per inch; balanced input provided to *y* amplifier; blue-grey metal cabinet with leather carrying handle.

For more data circle MD 46, Page 193

DATA ANALYSIS UNIT 48

. . . facilitates handling of graphic data

Telecomputing Corp., 133 *E. Santa Anita Ave., Burbank, Calif.*



Graphic image is projected on glass screen and measured by manually operating handwheels to line up cross wires with desired points.

Designation: Universal Telereader.

Size: 36 in. wide, 36 in. deep, 56 in. high; wt., 500 lb; screen size, 12 x 24 in.; measures records 16 mm-12 in. wide, to 100 ft. in length.

Service: Measures distances on all types of translucent or opaque records; direct-reading illuminated scales on cross-wire handwheels furnish reference co-ordinates; measurements also in electrical form for automatic tabulation or computation; record projection magnifications of 2x, 4x, 11x with reading accuracies of 0.003, 0.001, 0.0004-in. respectively; adjustable record motion, ½ ipm-100 fpm forward, 10-100 fpm reverse; hand and foot controls located for minimum fatigue; operates in normally lighted room; power requirement, 105-125 v, 60 cycles, 800 w.

Design: Console cabinet; sloping projection screen of ground glass; dual illumination with mercury vapor lamps for opaque records, fluorescent lamps for transparent records; cooling blowers; two orthogonal stainless steel cross-wires mounted on independent pulley systems; aluminum; digital recorders and punched-card equipment available.

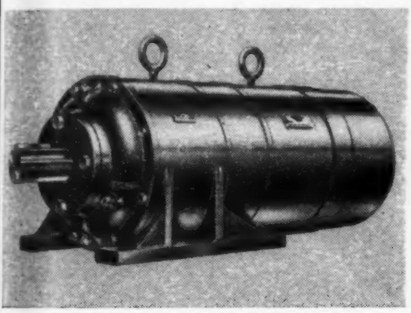
For more data circle MD 48, Page 193

Advertisement

Unit Assemblies of Motor, Speed Reducer and Brake Simplify Designs

Designers of motor-driven equipment which must provide for both speed reduction and a means of stopping the machine rapidly, find that it is frequently possible to simplify their product designs through the use of integral, compact assemblies of motor, speed reducer and brake.

Space-saving is an important benefit, since the integral assembly is considerably more compact than three separate units. Moreover, it is unnecessary for the designer to provide for means of coupling or assembling the units.



Typical Gear-Brakemotor design, showing the compactness of the assembly

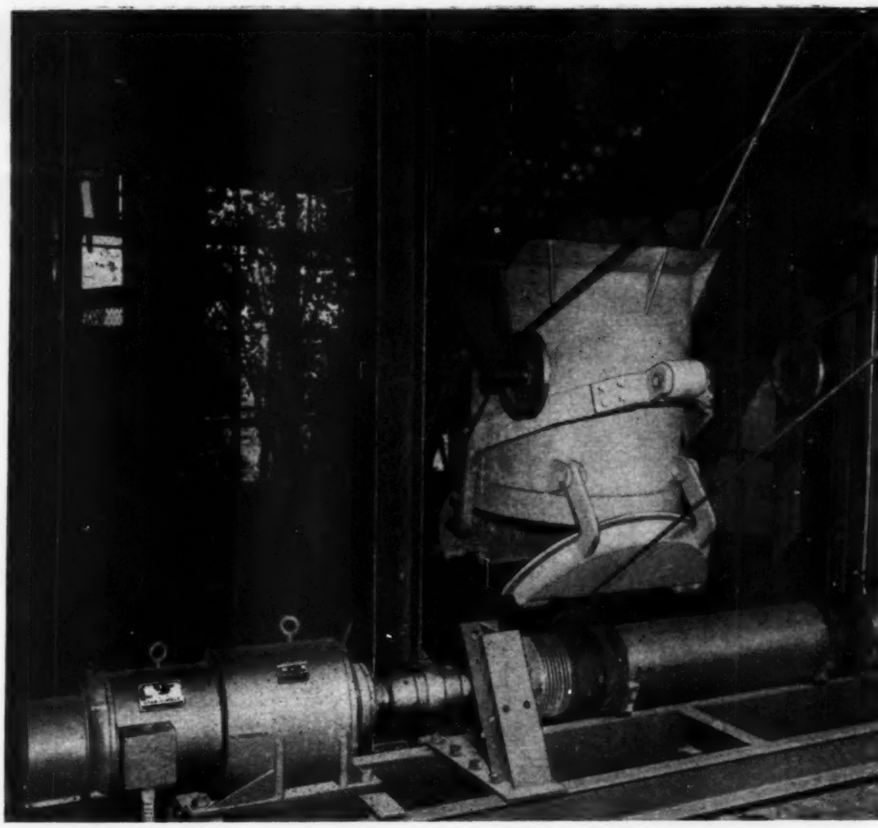
Adaptability to Varying Requirements

While the complete Gear-Brakemotor is a unit assembly, a very wide latitude is possible in selecting the individual components. Star-Kimble, the manufacturer of this type of motor, is prepared to supply a considerable range of motor ratings, braking torques and speed ratios.

It is the practice of many product designers merely to furnish Star-Kimble with the details of service requirements. Star-Kimble engineers then design the complete unit to meet these requirements, with all parts built to work together smoothly and efficiently. From the standpoint of the product designer, this procedure has the advantages of simplifying his own calculations, and also of establishing unified responsibility for the functioning of the complete unit.

The braking unit used in these Gear-Brakemotors is the same design employed in Star-Kimble Brakemotors supplied without the speed reducer feature. These Brakemotors have consistently proved their merits in every type of service calling for fast stops and starts on a heavy duty cycle.

Further information is contained in Bulletin B-501-A, available from Star-Kimble Motor Division of Miehle Printing Press and Mfg. Co., 201 Bloomfield Avenue, Bloomfield, New Jersey.



"YEARS OF LIFE IN TOUGHEST FOUNDRY SERVICE"

. . . say Whiting Engineers of Star-Kimble Gear-Brakemotors

The Whiting Skip Charger shown in the photo works two shifts a day, handling heavy charges of metal, coke and stone for the cupola. The Skip Charger must stop the load *accurately* and hold it *positively*. Equipment is subjected to intense heat and heavy concentrations of dirt in the air.

For tough service like that, Whiting powers its equipment with Star-Kimble Gear-Brakemotors, which have *proved* their ability to give years of service with little need for maintenance attention. The quick-acting, positive brake requires little or no adjustment; the motor and speed reducer provide a high factor of safety for heavy overloads. AND . . . motor, brake and speed reducer are a *single*, compact, well-protected unit.

For information on Star-Kimble Brakemotors, with or without the speed-reducing feature, write for Bulletin B-501-A.

STAR-KIMBLE

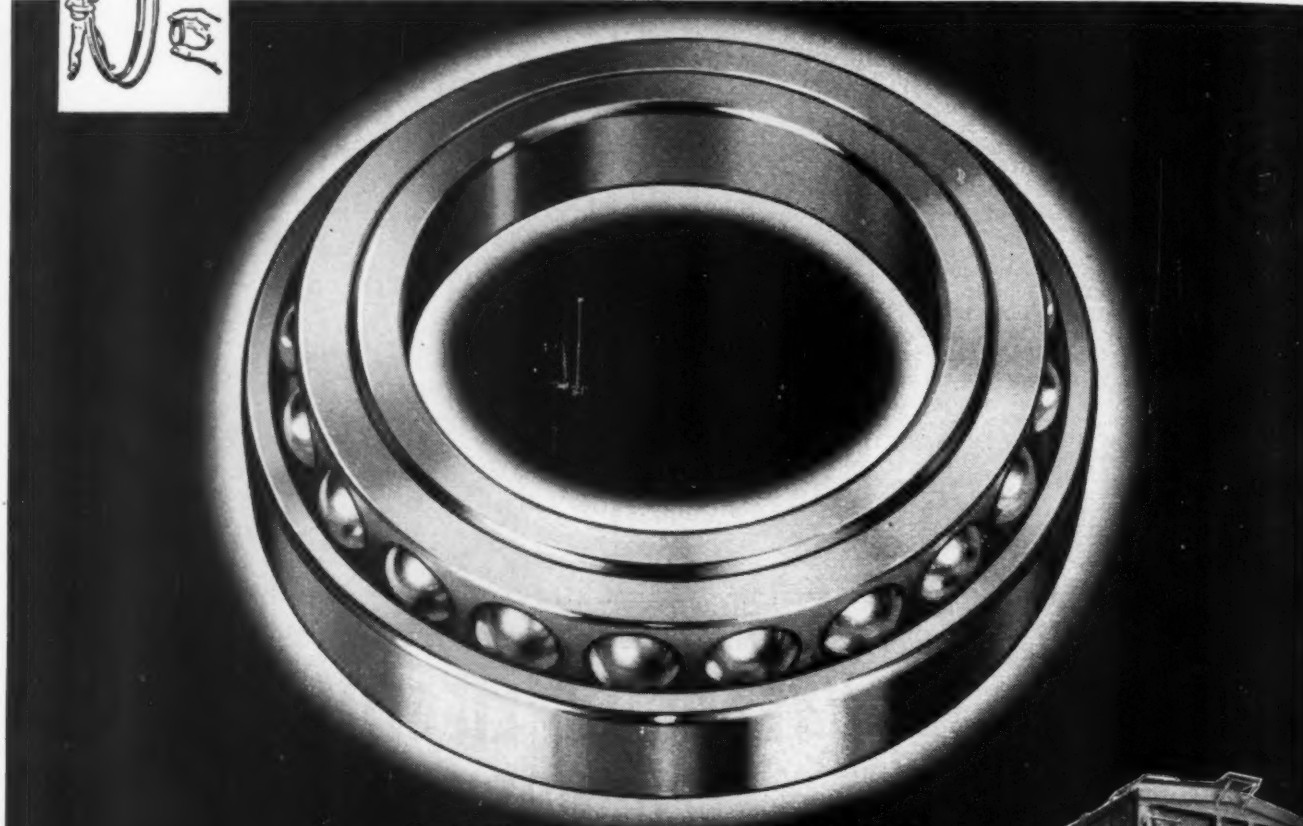
MOTOR DIVISION

MIEHLE PRINTING PRESS AND MFG. CO.

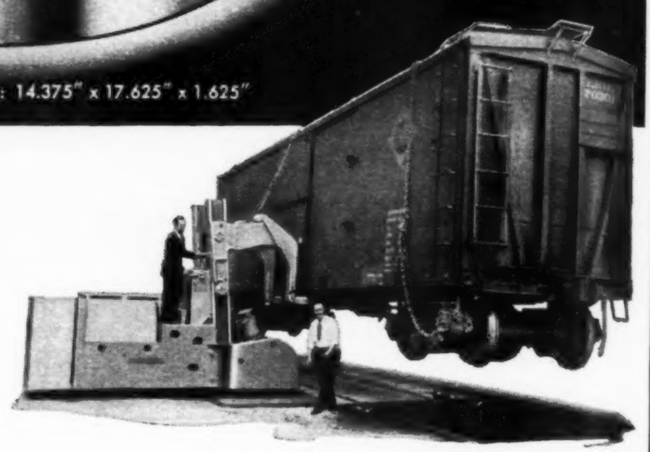
201 Bloomfield Avenue Bloomfield, New Jersey

Contact KAYDON of Muskegon

FOR ALL TYPES OF BALL AND ROLLER BEARINGS: 4" BORE TO 120" OUTSIDE DIAMETER



Special KAYDON Ball Bearings: 14.375" x 17.625" x 1.625"



KAYDON-bearinged Lift Trucks pick up 30-ton freight cars

HUSKY KAYDON Special Ball Bearings . . . with static thrust capacity of 120,500 lbs. each . . . help Automatic Skylift Giant Electric Trucks lift multi-ton loads smoothly, efficiently. Made by Automatic Transportation Company, these KAYDON-bearinged trucks live up to their claim to "lighten life's loads." They shoulder heavy responsibilities without flinching!

The precision of these bearings is vital to the steering ease of these powerful lift trucks. They support the heavily loaded trail axle wheels, for steering ease. Diametrical ball clearance is closely held. These special ball bearings are typical of the unusual designs KAYDON engineers create to help machine designers achieve their objectives.

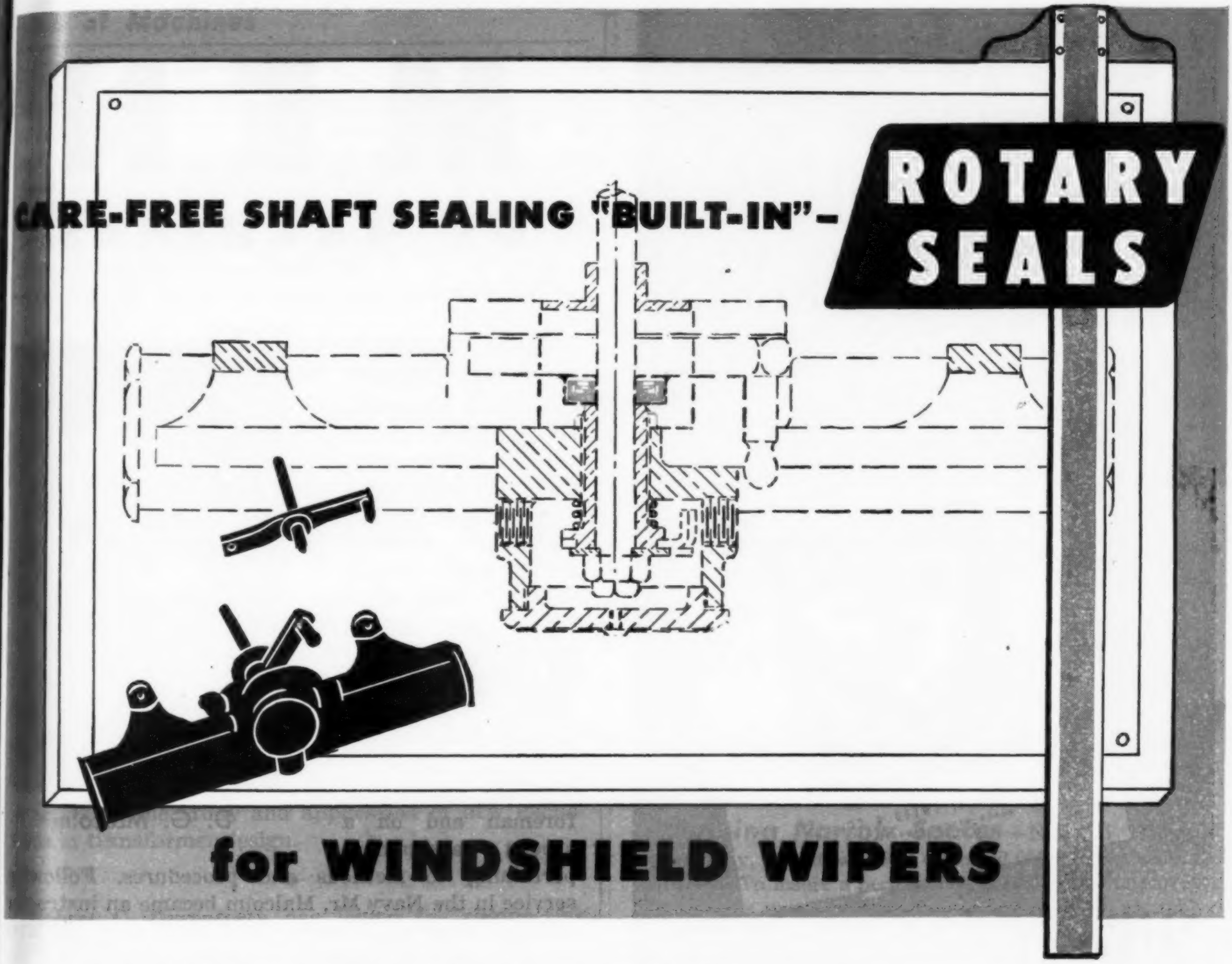
For dependable counsel on Precision Bearings and Needle Rollers, contact KAYDON of Muskegon.

KAYDON Types of Standard and Special Bearings:
Spherical Roller • Taper Roller • Ball Radial • Ball Thrust • Roller Radial • Roller Thrust • Bi-Angular Bearings

THE KAYDON ENGINEERING CORP.

MUSKEGON • MICHIGAN

PRECISION BALL AND ROLLER BEARINGS



A diminutive size is sometimes deceptive—as in the dimensions of many interesting special adaptations of the ROTARY SEAL principle. As an integral part of a windshield wiper motor, for example, the Rotary Seal serves the important function of an air trap under the hard continuous use to which the device is put.

Years of actual use have demonstrated that ROTARY SEALS are the best answer for difficult Shaft Sealing jobs of every nature. Whether the specific application calls for an unusually small unit such as this, or a much larger and more elaborate design, our engineers can draw on exceptionally broad experience to develop the exact answer.

THE ROTARY SEAL PRINCIPLE

is the original approach to a practical solution of a universally troublesome problem. Our booklet "SEALING WITH CERTAINTY" explains and illustrates the principle. We're glad to send it to you without obligation.

Why not check on the possibilities of ROTARY SEALS for greater *Certainty in Shaft Sealing* in *your* equipment? Consultation at the drawing board stage often helps make design easier, more practical.

THE ROTARY SEAL COMPANY
mechanical seals for rotating shafts

2022 NORTH LARRABEE STREET
CHICAGO 14, ILLINOIS, U. S. A.

MEN OF MACHINES

D. G. Malcolm

Bacon Vulcanizer Mfg. Co., Oakland, Calif., has appointed **D. G. Malcolm** to the position of vice president of engineering and production. Mr. Malcolm graduated from Purdue University in 1940 and joined the Allison Div. of General Motors Corp. in Indianapolis, where he served as calibration engineer, test stand engineer, production foreman and on a special assignment pertaining to methods and procedures. Following service in the Navy Mr. Malcolm became an instructor in engineering mathematics at Purdue and was later named assistant professor in this subject. He obtained his master's degree in industrial engineering at this time. In 1948 he joined the staff of the University of California as assistant professor of mechanical engineering and since then has served as general chairman, Industrial Engineering Institutes; president, Society of Industrial Engineers; and vice president of the western region of the American Institute of Industrial Engineers. He is a member of Tau Beta Pi, Sigma Xi, the American Society of Mechanical Engineers, the Society for the Advancement of Management, and the American Society for Engineering Education.

♦

Lincoln Engineering Co., St. Louis, has announced the appointment of **Carl H. Mueller** to the position of director of engineering. Formerly assistant to the president in charge of product development, Mr. Mueller will now assume full charge of engineering and research.

♦

Associate editor of MACHINE DESIGN for the past year, **Elman R. Dunn** has joined the Gardner Machine Co., Beloit, Wis., where he will be in charge of research and development work on a long-range project of redesigning the company's line of grinding machines. Mr. Dunn served an apprenticeship and later was assistant machine shop foreman for the Cumberland and Pennsylvania Railroad Co. at Mt. Sav-

age, Md. He then joined the engineering department of Landis Tool Co. in Waynesboro, Pa., where he worked as a designer, conducted experimental and research projects, assisted field service engineers and wrote service manuals. After leaving Landis, Mr. Dunn was chief engineer of Altens Foundry and Machine Works Inc. in Lancaster, O.

♦

William L. Ringland was recently named chief engineer, motor and generator section of the power department of Allis-Chalmers Mfg. Co., Milwaukee. Mr. Ringland joined the company in 1935 and, after completing the graduate training course, was assigned to the section which he now serves in his new capacity. Other appointments in this section announced at the same time include that of **J. F. Sellers** as assistant chief engineer, **L. T. Rosenberg** as engineer in charge of ac design, and **Chester Weilbaecher** as engineer in charge of dc design.

New assistant chief engineer at Allis-Chalmers' Pittsburgh Works is **William M. Terry Jr.**, who has been active in promoting the power transformer design program at this plant for approximately three years. Mr. Terry fostered the development of the company's Class H sealed dry type transformers and is active in the study and application of new insulations in transformer design.

♦

Robert C. Dyrenforth has joined the engineering staff of The Parker Appliance Co., Cleveland. For seven years Mr. Dyrenforth served as assistant chief engineer for the Swartwout Co. and was also associated with the Bailey Meter Co. for eight years.

♦

W. C. Jones has been appointed chief engineer at the Bridgeport, Conn., works of Underwood Corp. Formerly chief product engineer in the company's Hartford, Conn., plant, he will now be in charge of the engineering of accounting and adding machines. Replacing Mr. Jones in Hartford is **W. J. Gove**, who has worked in the computing machine and electric typewriter departments, tool designing, planning department, and plant layout and in the standards engineering department. Prior to his present appointment, Mr. Gove served for five years as quality engineer.

♦

Five engineers recently received promotions at Worthington Corp., Harrison, N. J. **John E. Lancaster** was appointed assistant chief engineer of the air conditioning and refrigeration engineering division. He joined the company in 1940 as a test assistant in the experimental test department and since then has served in the research and development department as a test engineer, group leader, assistant manager and acting manager. **William C. Osborne** has been appointed manager of the research and development department. He was formerly associated with the centrifugal engineering department as a re-

A DIGEST OF NEW DEVELOPMENTS IN HIGH PERFORMANCE PUMPS

PISTON TYPE FLUID POWER PUMPS

Constant and Variable Delivery Types . . . 0.25 gpm to 10 gpm . . . direct engine driven and motorized units.

Featuring continuous working pressures to 3000 psi and continuous speeds to 3750 rpm, today's STRATOPOWER oil hydraulic Pumps (both constant and variable delivery types), afford special advantages for aircraft and other applications, The dual pressure Pumps incorporate remote oil pilot controlled pressure regulator. An electric modification provides selective pressure control and Pump unloading. These Pumps are self-priming and develop suction line pressures approaching 1" Hg. absolute and will also operate under conditions of high reservoir pressurization.

VANE TYPE FLUID POWER PUMPS

Capacities 3 gpm to 120 gpm. Standard models rated at 1200 rpm . . . specials to 3600 rpm.

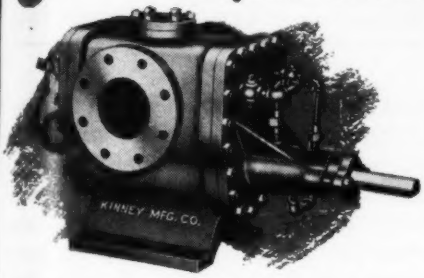
A distinctive Dual Vane construction, which provides increased fluid delivery rates (even with extremely thin fluids), makes DUDCO Hydraulic Pumps and Motors the first single-stage vane type proven for 2000 psi operation. The minimum size and high efficiencies of these Pumps and Motors create new opportunities in the design of hydraulic fluid power systems for all types of industrial equipment as well as countless applications on heavy-duty machinery and ordnance vehicles.

GEAR TYPE FLUID POWER PUMPS

½ gpm to 130 gpm for working pressures to 1500 psi and operating speeds to 2000 rpm.

A unique Four-Bolt design, which locates the assembly bolts within the area of greatest internal pump pressure, indorses HYDRECO Hydraulic Pumps for the heavy-duty required in equipment for the construction and materials handling fields. This Four-Bolt design provides the rigidity and stability that reduces distortion of housing parts and wear plates and insures against uneven wear and loss of overall efficiency in the face of extreme mechanical and hydraulic loads.

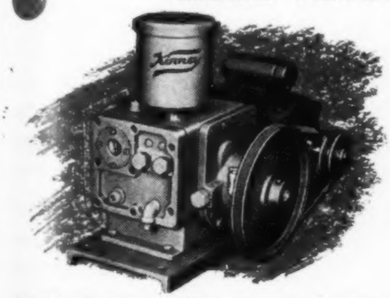
LIQUID HANDLING PUMPS



Plain or steam jacketed Rotating Plunger or Heliquad Types 2 gpm to 3360 gpm.

Virtually any material that will flow through a pipe, including difficult viscous fluid, can be handled with meter-like volumetric accuracy with today's KINNEY Rotary Plunger and Wide Angle Herringbone Gear Pumps. The Rotary Plunger Pump features a construction with no valves, blades, pistons or springs. The versatile line of Herringbone Gear Pumps includes models driven by timing gears with anti-friction bearings located outside the pump chamber. Both types available with or without heating jackets.

HIGH VACUUM PUMPS



Single Stage and Compound types, ¼ HP at 2 cu. ft. per min. to 75 HP at 1800 cu. ft. per min.

There is only one principle which has been found suitable for Vacuum Pumps in all capacity ranges . . . that of the Rotary Plunger employed in KINNEY High Vacuum Pumps. First to use the oil-sealed Rotary Plunger, these Pumps develop absolute pressure readings of 0.1 Micron (0.0001 mm Hg.) or better. Alone or in combination with oil diffusion Pumps, they provide the answer to the most exacting high vacuum applications in the electronic, processing and research fields.

MOTORS, VALVES & CYLINDERS

In addition to the complete range of Pumps described, there are equally important components . . . Hydraulic Motors, Valves and Cylinders . . . all available from a single source. DUDCO Hydraulic Motors, employing the remarkably efficient DUAL-VANE principle, with high running torques averaging 90% or more of theoretical at any speed down to nearly stalled and with smooth operation under load. Models rated from 9 to 720 in. lbs./100 psi for 2000 psi operation. HYDRECO Hollow Plunger Valves in single or multiple plunger units for controlling single, double-acting or telescopic HYDRECO Cylinders as well as other Hydraulic Power Units . . . capacities from ½ gpm to 150 gpm and for operating pressures to 1500 psi. Relief Valves, Pressure Regulators, Flow Dividers and other special purpose Valves are available for nearly any type of Hydraulic circuit.

The New York Air Brake Company and its affiliates provide a most comprehensive coverage of Pumps and related equipment for the needs of defense and industry. Here, in one organization, is "Know How" teamed with advanced facilities and a tradition of precision and craftsmanship. Here is research and development dedicated to the constant improvement and the ever-broadening service which hydraulic and vacuum equipment can contribute now and in the future.

Catalogs and complete information on the Hydraulic Pumps, Motors and other components herein described are available on request.

THE NEW YORK AIR BRAKE COMPANY

420 LEXINGTON AVE., NEW YORK 17

AFFILIATES: DUDCO DIVISION • • • HYDRECO DIVISION • • • KINNEY MANUFACTURING CO.

search engineer and design engineer. **Norman L. Meyerson,** who joined Worthington's research and development department in 1942, is now assistant manager of that department. Having served as test engineer and junior engineer in the experimental test, centrifugal engineering and research and development departments, **Frederick C. Gilman** has been appointed research engineer. **T. A. Herman** has been named assistant chief engineer of the reciprocating engineering division. He joined the company as a test engineer in 1937 and became a design engineer in 1941.

♦

Martin A. Edwards has been named manager of engineering of the X-ray department of General Electric Co. in Milwaukee. Dr. Edwards, who has been engineering manager of the company's general engineering laboratory in Schenectady, N. Y., succeeds **William J. Fleming,** whose appointment as general manager of the lighting and rectifier department in Lynn, Mass., was announced recently.

♦

Appointment of **R. R. Harrison** as assistant chief engineer and **Bruce MacDonald** and **B. E. McNay** as senior project engineers was announced recently by The Rucker Co., Oakland, Calif.

♦

Newly appointed to the engineering division at Midwest Research Institute, Kansas City, Mo., are **Richard Fetter,** electronics research engineer; **Wickliffe B. Hendry** and **Samuel Zivi,** senior mechanical engineers; **Robert Miller,** senior research mechanical engineer; and **Walter Vorland Jr.,** mechanical engineer.

♦

T. H. Wickenden, vice president in charge of development and research of The International Nickel Co. Inc., was recently elected a member of the Welding Research Council of the Engineering Foundation for a three-year term.

♦

In expanding its research activities Aerovox Corp., New Bedford, Mass., has named **Henry Taylor** as assistant to the director of research and **Peter P. Grad** as technical director. **Rex Nicholson** has been appointed research engineer, and **Lenine Gonsalves** has joined the research department in the capacity of electrical engineer.

♦

Chief engineer in charge of research and engineering since 1946, **F. C. Messaros** has been appointed vice president in charge of engineering by American Engineering Co., Philadelphia.

♦

P. R. Mallory & Co. Inc., Indianapolis, has announced the appointment of **Victor Welge** as associate director of engineering. Mr. Welge formerly headed the staff of the electronics and missile section of Consolidated Vultee Aircraft Corp., San Diego, Calif. In

his new position he will be concerned with electric, electronic and mechanical engineering problems in the central engineering department and will assist in all administrative duties concerning engineering activities in the firm's ten manufacturing divisions.

♦

Formerly special design engineer of the engine division of the National Supply Co. at Springfied, O., **Vernon L. Durrstein** has been appointed assistant chief engineer of the division.

♦

Nesco Inc., Milwaukee, has announced the appointment of **Cecil J. Schanz** as chief engineer of methods and standards for all plants of the company.

♦

Robert E. Sheahan recently joined the engineering staff of The Parker Appliance Co., Cleveland, as product development engineer. Mr. Sheahan has been associated with General Electric Co. in Cleveland and Bridgeport, Conn., and for the past ten years has served as design engineer for a variety of projects involving both GE products and military equipment.

♦

Chief engineer since 1946, **John A. Cortelli** has been elected to assume the additional duties of vice president of engineering for Clark Controller Co., Cleveland.

♦

After serving for eleven years as head of the research and development laboratories of Norton Co., Worcester, Mass., **Milton F. Beecher** has retired as vice president and director. **Wallace L. Howe**, director of research and development, succeeds Mr. Beecher.

♦

The New Departure Div. of General Motors Corp., Bristol, Conn., has reorganized its engineering department in order to broaden activities associated with research, engineering, development and application of instrument ball bearings. A special section has been formed which will be under the direction of **Kenneth D. Mackenzie,** assistant chief engineer. Mr. Mackenzie was formerly assistant plant manager of the division's operations at Meriden, Conn. **Raymond J. Lynch,** also an assistant chief engineer, will be in charge of all other bearing applications.

♦

John L. Young, vice president in charge of engineering for United States Steel Co., has been elected president of the Association of Iron and Steel Engineers for 1953.

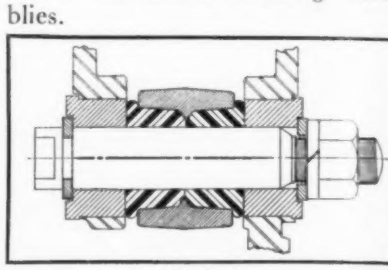
♦

Chief engineer of Stevens Mfg. Co. Inc., Mansfield, O., since 1950, **Charles E. Mertler** has been named vice president in charge of engineering and development for the company.

Advertisement

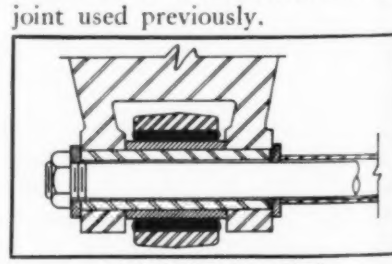
Lord Bonded Rubber Improves Radius Rod Joint on Trailmobile

Trailmobile, Inc. at Cincinnati, Ohio, believed that maintenance costs on their suspension systems could be reduced by redesign of the radius rod rubber bushings. When their standard radius rod had to be disconnected for maintenance of the suspension, often it was necessary to replace a number of parts of the radius rod rubber bushing assemblies.



Old Design

Lord engineers worked with the designers at Trailmobile to arrive at a rubber joint design that permits making an almost limitless number of disconnections of the radius rods without requiring replacements of the rubber joints, requiring only the time needed to remove the pin through the joint and bracket. The Lord Center-Bonded Joint, now standard in new Trailmobile assemblies, not only fully satisfied requirements, but did so with fewer and simpler parts, *and* at a cost lower than that of the unbonded rubber joint used previously.



Lord Design Now Used

Now when it becomes necessary to remove the Trailmobile Radius Rod it is a simple, easy operation to drive the pin from the hanger. When new bushings are necessary the old ones are pressed out and new ones installed with an arbor press.

Improvements of this nature are being designed into a wide diversity of automotive and industrial products as the result of cooperative efforts of Lord Engineers and the Design Engineers of such product manufacturers. Precision manufacture is the added ingredient which gives Lord Engineering its high value to industrial designers.

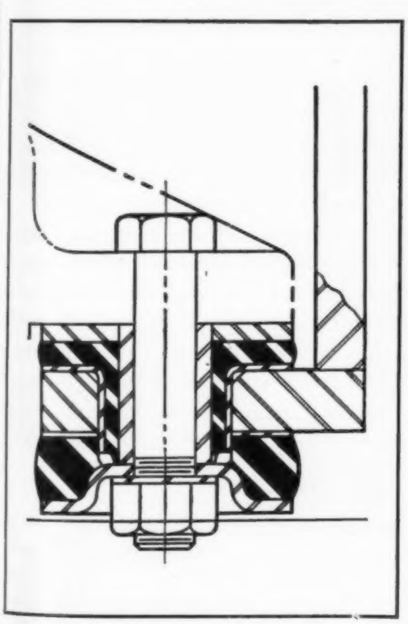
Advertisement

Lord Mountings Cut Service Costs on Ready-Power Units Used by Industry

The most efficient mounting system for Ready-Power "Live Power" units necessitates the isolation of engine vibration, the accommodation of frame twist and absorption of shock loads encountered in industrial materials handling.

Because of limited space the mountings must be sufficiently rigid to prevent excessive engine-generator motion yet flexible enough to isolate engine vibration and absorb shock loads which seriously damage engine brackets, oil lines and other component parts. These shock loads are caused when industrial trucks bump into pallets or solidly packed storage areas or are operated over unusually rough and bumpy surfaces.

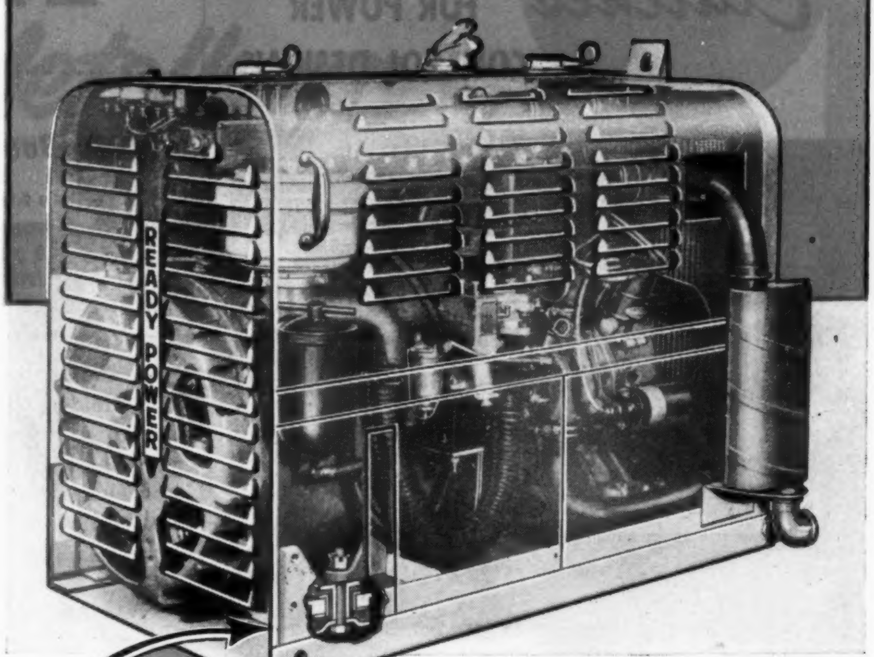
DuPont Neoprene is used as the base in the compound specifically designed by Lord to meet the deteriorating effects of oil, gasoline etc.

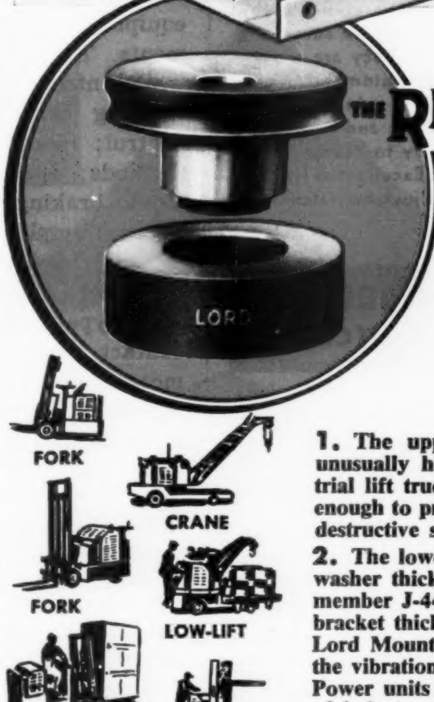


This Lord mounting is designed for easy installation, no extremely close tolerances being involved. The Ready-Power representatives maintain close contact with their equipment under all operating conditions. They report that the use of Lord mountings minimizes maintenance and service formerly the result of shock and vibration.

Further details on this and other successful handling of vibration and shock problems are available on request from Lord Manufacturing Company, Erie, Pa.

LORD MOUNTINGS
SERVE DUAL PURPOSE





on

THE READY-POWER CO.

"Live Power Units"

LORD Shock Mountings accomplish two vital objectives in the delivery of "Live Power" generated as needed directly on the truck chassis of industrial fork trucks, tractors, cranes and locomotives by Ready Power Units.

1. The upper Lord Mounting J-4497-2 absorbs the unusually high "g" shock loads encountered in industrial lift truck service . . . At the same time it is rigid enough to prevent excessive engine motion due to these destructive shock loads.

2. The lower member J-4591-1 is a rebound snubbing washer thicker than the sandwich section of the upper member J-4497-2. Precompression thus allows variable bracket thickness of plus or minus 1/16 inch. Thus the Lord Mountings serve the dual purpose of minimizing the vibration and the multiple shocks to which Ready-Power units are subjected in powering the heavy tools of industry. You can profit by Lord experience in the control of vibration and shock. Write or call . . .

BURBANK, CALIFORNIA 233 South Third Street	DALLAS, TEXAS 1613 Tower Petroleum Building	PHILADELPHIA 7, PENNSYLVANIA 725 Widener Building	DAYTON 2, OHIO 238 Lafayette Street
DETROIT 2, MICHIGAN 7310 Woodward Ave.	NEW YORK 16, NEW YORK 280 Madison Avenue	CHICAGO 11, ILLINOIS 520 N. Michigan Ave.	ERIE, PENNSYLVANIA 1635 West 12th Street

LORD MANUFACTURING COMPANY • ERIE, PA.

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Headquarters for
VIBRATION CONTROL

THE ENGINEER'S Library

Recent Books

Controllers for Electric Motors. *By Henry Duvall James, consulting engineer, and Lewis Edwin Markle, design engineer; 440 pages, 6 by 9 inches, clothbound; published by McGraw-Hill Book Co. Inc., New York; available from* MACHINE DESIGN, $7.00 *postpaid.*

Written as a basic guide on the selection, operation and maintenance of control equipment for electric motors, this second edition includes new developments subsequent to the publication of the original version. Involved mathematics have been avoided and elementary chapters have been included for those unfamiliar with this type of electrical apparatus. Types and methods of control discussed are confined to the generally available commercial components.

Text material begins with fundamental problems of control and proceeds through the various types of equipment concluding with latest control developments. Topics covered are function and types of control; controller diagrams; magnetic contactors; motor starting characteristics with different methods of control; motor acceleration methods; speed control methods; special applications; mechanical and dynamic braking; regeneration; voltage control for dc motors; amplifiers for motor control; magnetic amplifiers; series-parallel control and the electropneumatic contactor; adjustable speed ac motors of the wound-rotor type; resistors; manual controllers; dc magnetic contact controllers; ac controllers; synchronous-motor control; electron-tube control; remote and supervisory control; protective devices; codes, installation and maintenance; and control developments.

Mechanics of Vibration. *By H. M. Hansen, professor of engineering mechanics, and Paul F. Chenea, associate professor of engineering mechanics, University of Michigan; 431 pages, 5¾ by 9 inches, clothbound; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN, $8.00 *postpaid.*

Basic principles of the theory of mechanical vibrations and their application to engineering problems are treated in this volume which has been designed primarily as a textbook for both undergraduate and graduate students. Specific applications and problems have been minimized; main emphasis has been placed on theory and general problem techniques. An appendix to the main text contains problems graded in difficulty for each chapter, many with answers.

Introductory material on general concepts precedes the main portion of the text which is divided into three parts. Part 1 deals with systems of one degree of freedom and covers free vibration without damp-

ing, forced vibrations without damping, and forced vibrations with damping. Part 2 extends the theory to systems of several degrees of freedom and discusses the classical method; the mobility method and use of the complex variable; and solution of the general frequency equation. Part 3 consists of introductory information on special topics which are necessary for a more general and refined analysis of vibration problems. Topics presented cover systems with distributed physical constants, vibrations of a transient character and vibrations of a nonlinear character.

Practical Descriptive Geometry. *By Hiram E. Grant, professor and head, department of engineering drawing, Washington University;* 261 *pages,* 6 *by* 9 *inches, clothbound; published by McGraw-Hill Book Co. Inc., New York; available from* MACHINE DESIGN, $4.00 *postpaid.*

This descriptive geometry textbook, according to the author, "attempts to correlate and integrate theory and practice instead of considering each a separate entity." The "direct method" of presentation, which is a comparatively recent American innovation, is employed throughout in this treatment of the subject. Problems have not been included in the text but a special set designed for use with the book is available.

Farm Gas Engines and Tractors. *By Fred R. Jones, professor and head of agricultural engineering, Agricultural and Mechanical College of Texas;* 449 *pages,* 6 *by* 9 *inches, clothbound; published by McGraw-Hill Book Co. Inc., New York; available from* MACHINE DESIGN, $6.00 *postpaid.*

A third edition of the original 1932 version, this textbook deals with farm power—the internal combustion engine as a stationary unit and the tractor as an automotive unit. All information has been brought up to date and a chapter on thermodynamics has been added.

Manufacturers' Publications

Aluminum Forming. 151 *pages,* 5¾ *by* 8¾ *inches, ring bound; available on company letterhead request from Reynolds Metals Co., 2500 South Third St., Louisville* 1, *Ky.*

Although aluminum is an easy metal to form, the basic techniques necessary for its satisfactory fabrication will often differ from those normally applied to other metals. This manual deals with the important factors concerning aluminum metallurgy and fabrication. Text material is divided into four sections: Section 1 is devoted to metallurgical and technical data pertaining to sheet and plate; Section 2 covers the various types of sheet forming operations; Section 3 presents data on tubing and pipe; and Section 4 discusses forming, bending and swaging of tubing and pipe. Well illustrated with draw-

ings and photographs, the manual also contains many helpful tables and formulas.

Practical Metallurgy for Engineers. 603 *pages,* 5½ *by* 8½ *inches, clothbound; available from E. F. Houghton & Co.,* 303 *West Lehigh Ave., Philadelphia, Pa.;* $3.50.

Now in its fifth edition, this handbook covers the latest standards and practices of the metal industry as well as future probabilities for new metals, alloying elements and types of heat treatment. In line with the company's experience, the sections of the book covering heat treating make up the major portion of the text material. Compared to the previous 1943 edition, this volume has been enlarged by 120 pages. New chapters include one on flame and induction heating and one on surface-hardening methods other than carburizing. Additionally, a discussion of the heat treatment of magnesium and copper alloys has been included.

Government Publications

Mechanical Properties of Metals at Low Temperatures. *National Bureau of Standards Circular* 520; 210 *pages,* 5¾ *by* 9 *inches, clothbound; available from Government Printing office, Washington* 25, *D. C.,* $1.50 *postpaid.*

Subject matter for this book consists of nine papers presented at the National Bureau of Standards 1951 symposium on the influence of low temperatures on the mechanical properties of metals. Topics covered include European work in the field; manufacture of steels for low-temperature service; development and application of chromium-copper-nickel steel; tensile properties of copper, nickel and some copper nickel alloys; aircraft applications; properties of austenitic stainless steels; dimensional effects in fracture; mechanical properties of high-purity iron-carbon alloys; and brittle fractures in ship plates.

NACA Technical Series. *Each publication is* 8 *by* 10½ *inches, paperbound, side-stapled; copies available from National Advisory Committee for Aeronautics,* 1924 *F St. N. W., Washington* 25, *D. C.*

The following Technical notes are available:

2671. **Fatigue and Static Tests of Flush-Riveted Joints**—38 pages

2716. **Effect of Open Circular Holes on Tensile Strength and Elongation of Sheet Specimens of a Magnesium Alloy**—24 pages

2717. **Effect of Temperatures from −70 to 600 F on Strength of Adhesive-Bonded Lap Shear Specimens of Clad 24S-T3 Aluminum Alloy and of Cotton and Glass-Fabric Plastic Laminates**—26 pages

2731. **Influence of Structure on Properties of Sintered Chromium Carbide**—21 pages

2737. **Plastic Stress-Strain Relations for Combined Tension and Compression**—61 pages

2745. **Influence of Chemical Composition on Rupture Test Properties at 1500 F of Forged Chromium-Cobalt-Nickel-Iron Base Alloys**—69 pages

2755. **Analysis of Landing-Gear Behavior**—98 pages

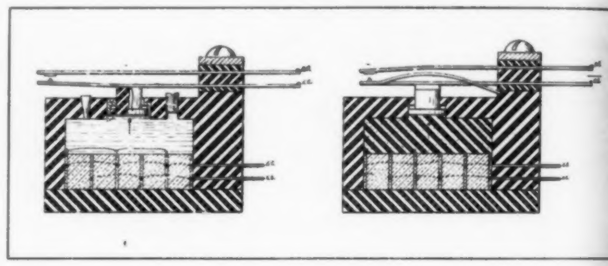
2758. **Wear and Sliding Friction Properties of Nickel Alloys Suited for Cages of High-Temperature Rolling-Contact Bearings. I—Alloys Retaining Mechanical Properties to 600 F**—30 pages

2759. **Wear and Sliding Friction Properties of Nickel Alloys Suited for Cages of High-Temperature Rolling-Contact Bearings. II—Alloys Retaining Mechanical Properties above 600 F**—29 pages

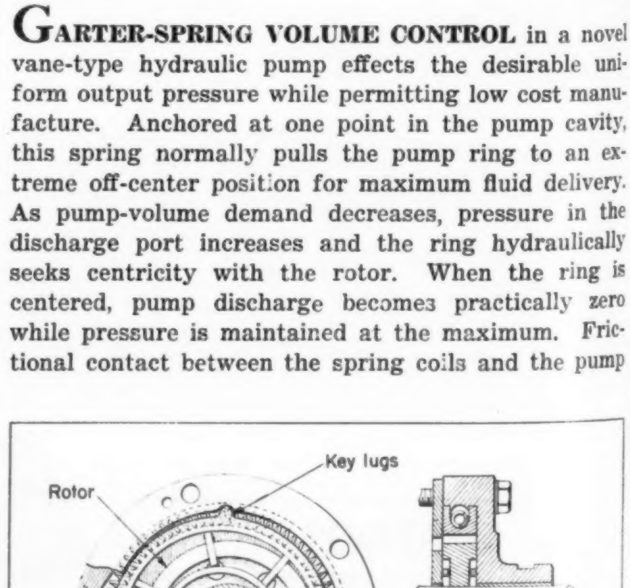
2767. **Dynamics of Mechanical Feedback-Type Hydraulic Servomotors Under Inertia Load**—63 pages

NOTEWORTHY Patents

HYDRAULIC AMPLIFICATION of switch contact motion in piezoelectric relays, instead of a mechanical lever amplification system, greatly simplifies this type of relay by reducing the number of parts previously necessary. Submerged in the hydraulic medium, the piezoelectric crystals deform when the control current potential is varied, displacing the fluid or semi-fluid accordingly. Proportionally amplified motion of a comparatively small output plunger results, actuating the movable contact finger of the relay. Patent 2,587,482 assigned to Bell Telephone Laboratories Inc. by Arthur C. Keller.



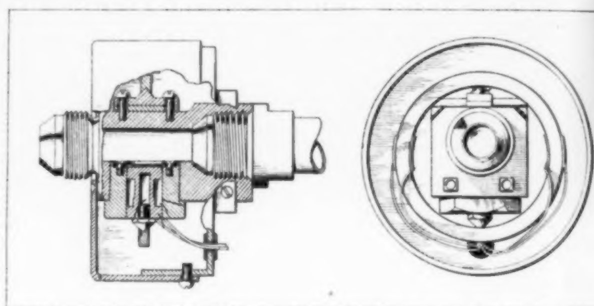
GARTER-SPRING VOLUME CONTROL in a novel vane-type hydraulic pump effects the desirable uniform output pressure while permitting low cost manufacture. Anchored at one point in the pump cavity, this spring normally pulls the pump ring to an extreme off-center position for maximum fluid delivery. As pump-volume demand decreases, pressure in the discharge port increases and the ring hydraulically seeks centricity with the rotor. When the ring is centered, pump discharge becomes practically zero while pressure is maintained at the maximum. Frictional contact between the spring coils and the pump



Noteworthy Patents

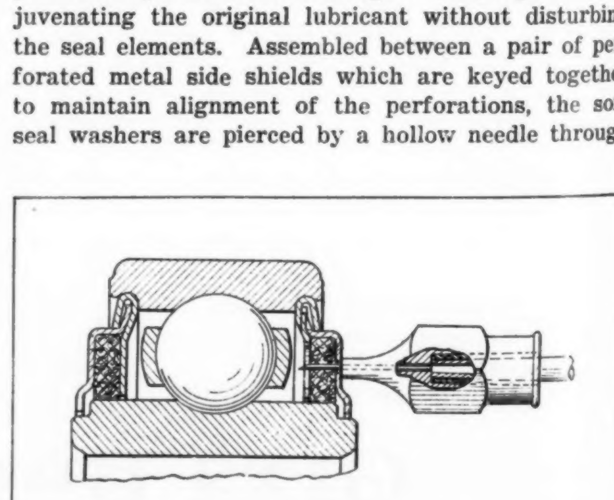
ring (groove) damps out critical vibrations resulti in smooth operation without hunting action. Pate 2,588,032 assigned to Packard Motor Car Co. James H. O'Brien.

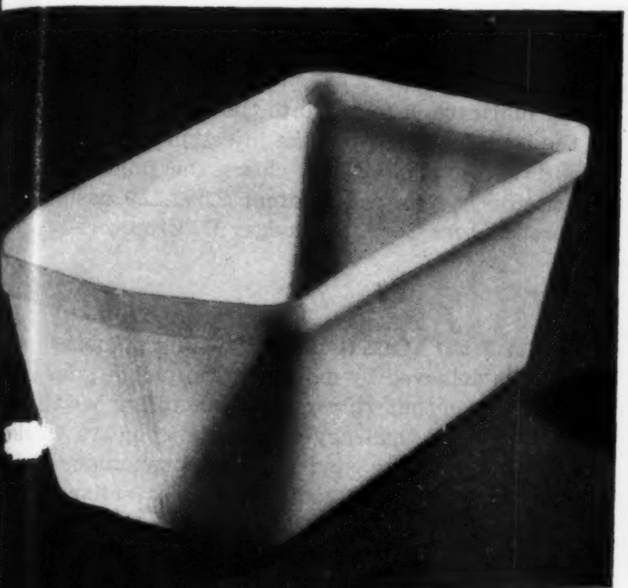
HYDRAULIC PRESSURE MEASUREMENTS high accuracy are possible with a new gage whi eliminates the fluid temperature sensitivity that ge erally affects results with direct electric strain gag readings. With pressure transmitted to an e proof ring through a sealed plunger, continuous rea

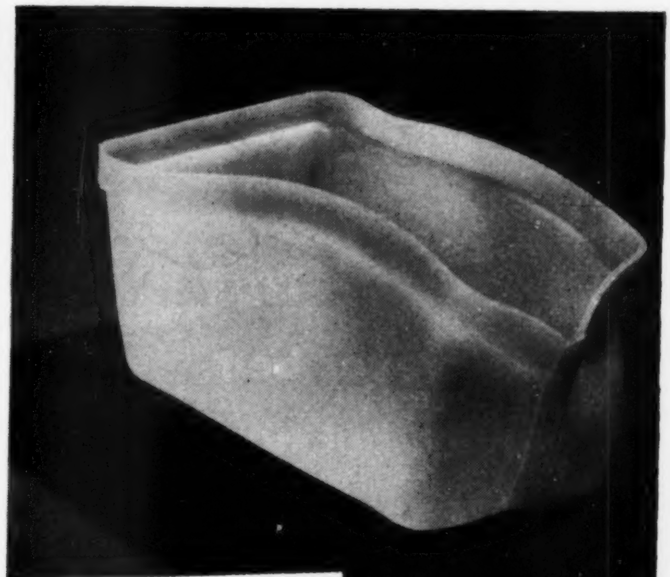


ings can be recorded without temperature-induced in accuracies. Strain gages are mounted diametricall opposed on innner and outer faces of the ring whic is pivot-mounted either on the side of or encirclin the gage body. Patent 2,585,350 assigned to Dougla Aircraft Co. by John D. Russell.

HYPODERMIC LUBRICATION prolongs the life o permanently sealed ball bearings in that it permits re juvenating the original lubricant without disturbin the seal elements. Assembled between a pair of per forated metal side shields which are keyed togethe to maintain alignment of the perforations, the sof seal washers are pierced by a hollow needle through







See the difference!

The boxes in these unretouched photographs were molded under the same conditions and exposed to a temperature of 212°F. The box on the left, made from Koppers MC 409, showed little or no distortion after 30 minutes at this temperature. The box on the right, made from a widely used commercial modified polystyrene, showed major distortion within 10 minutes.

high heat resistance in a modified polystyrene . . .

KOPPERS MC 409

• Here's the high-heat modified polystyrene that the industry has long wanted . . . Koppers MC 409.

This new material combines three important advantages . . . higher heat resistance *(up to 198°F.*)* . . . the toughness and shock resistance of modified polystyrene . . . and the desirable finish and molding characteristics of regular polystyrene. MC 409's excellent heat resistance is comparable to that of Koppers well known Polystyrene 8.

These qualities make MC 409 an ideal material for radio cabinets, battery cases, washing machine and refrigerator parts, toys, housewares, brush handles and, in fact, any molded or extruded polystyrene product subjected to heat.

Koppers MC 409 makes possible sturdy, shock resistant, plastic products that withstand the high temperatures found in show windows, dishwashers and automobiles, where other modified polystyrenes sag and shrink. Tests prove that on exposure to boiling water MC 409 will show low deformation with a minimum of shrinkage.

FOR FREE SAMPLES for testing in your laboratory and for Technical Bulletin C-2-161-TR describing the properties and characteristics of MC 409, write, wire or phone Koppers Company, Inc., Chemical Division, Dept. MD-112, Pittsburgh 19, Pa.

**A.S.T.M. D-648-45T.*

Koppers Plastics Make Many Products Better and Many Better Products Possible

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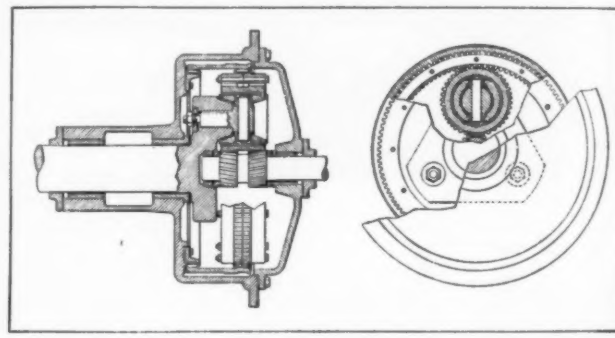
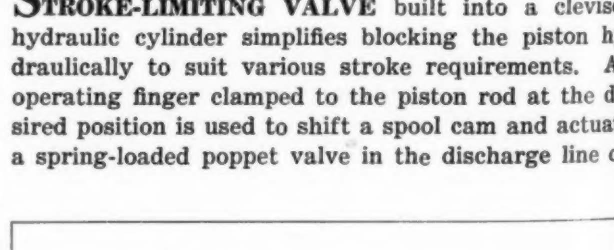
which lubricant is forced into the bearing. Resiliency of the seal washer wipes the injection needle clean as it enters the bearing and closes the puncture when the needle is removed. Patent 2,591,129 assigned to General Motors Corp. by Rodger D. Brouwer.

EQUALIZED TOOTH LOADING in a planetary gear reducer is achieved by mounting the planet gear bearings to the output spider studs on radial wrist pins. This avoids the tendency of the planet gears to skew because of normal spider deflections under heavy loads. Thus, uniform tooth loading is enhanced and



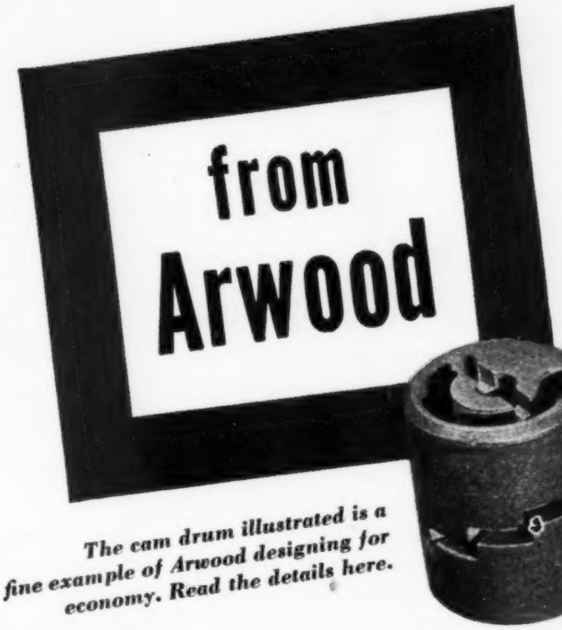
localized wear reduced. The internal ring gear of the reducer is supported by a splined coupling sleeve which permits it to shift axially and radially to a slight degree for further equalizing tooth loading and wear. Patent 2,591,734 assigned to General Electric Co. by Norman A. Smith and Kermit L. Darrah.

STROKE-LIMITING VALVE built into a clevised hydraulic cylinder simplifies blocking the piston hydraulically to suit various stroke requirements. An operating finger clamped to the piston rod at the desired position is used to shift a spool cam and actuate a spring-loaded poppet valve in the discharge line on



the pull stroke. Reversal of line pressure resets the piston during which time the stroke-control cam resets automatically. This arrangement eliminates the extra piping and valve gear otherwise required to provide similar stroke control. Patent 2,587,182 assigned to Bendix Aviation Corp. by Carlos B. Livers.

Precision casting news



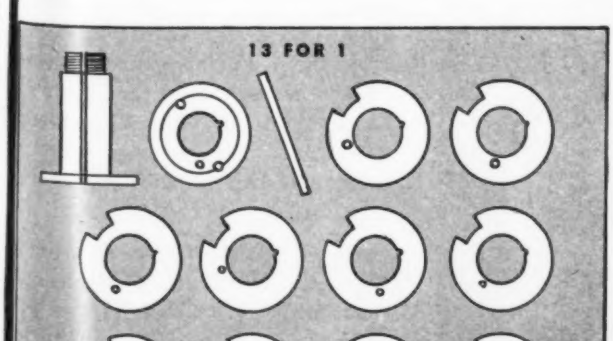
The cam drum illustrated is a fine example of Arwood designing for economy. Read the details here.

by Arwood Engineers

This is, indeed, interesting precision casting news. Study, for a moment, the cam drum illustrated above. Originally, it was made in 13 parts —a spindle — turned, drilled, slotted and threaded from bar stock — a cap, turned drilled in 4 places and tapped from bar stock — 10 cam segments, turned, milled and drilled from bar stock and a locking pin.

Arwood's engineers combined these 13 parts into 1 part — and eliminated all except 1 machining operation — something of a record. In addition the alloy was changed to give much better service and much longer life. Interesting to engineers are substantial savings that resulted.

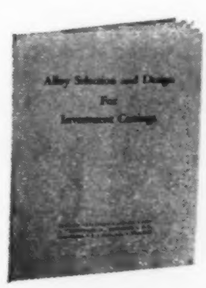
This is but one example of what Arwood's engineers are able to accomplish. They work in many industries and solve many problems. If you have a problem why not write Arwood — no obligation whatever.



LES GOTT

Les Gott has rejoined the Arwood Sales Organization as engineering representative in Connecticut, Westchester County, and the Hudson River Valley in New York State. His engineering background and long experience in investment castings make him able to provide competent engineering and metallurgical counsel. He is typical of Arwood men in the field.

Mr. Gott is a graduate of M.I.T., has been a Captain, Ordnance Corps, in charge of equipping and operating an investment casting foundry. Later he operated his own foundry. He was recalled to active duty in 1951 and upon his discharge recently he rejoined the Arwood Organization.



For more complete information write for our booklet, "Alloy Selection and Design for Investment Castings."

☆ ☆ ☆

An important new book—"Investment Casting for Engineers"—has just been published. It is highly recommended reading for all design engineers.

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Plants: Brooklyn, N.Y., Groton, Conn., Tilton, N.H.

Engineering News Roundup

Jet Engine Saves Weight, Space

Called the world's most powerful qualified turbojet aircraft engine, a new power plant has been developed and is being produced for the U. S. Navy. Capable of developing thrust equivalent to approximately 25,000 hp at present-day jet flight speeds, the new Westinghouse engine is said to be the first to provide constant speed drive for airplane accessories as an integral part of the engine. This feature will permit designers to make substantial savings in weight and space in new planes.

An outgrowth of the original Westinghouse J40 engine, the new model has successfully completed the Defense Department's 150-hour qualification test which all new engines must pass to be eligible for quantity production. The test was equivalent to more than 75,000 miles of actual flight.

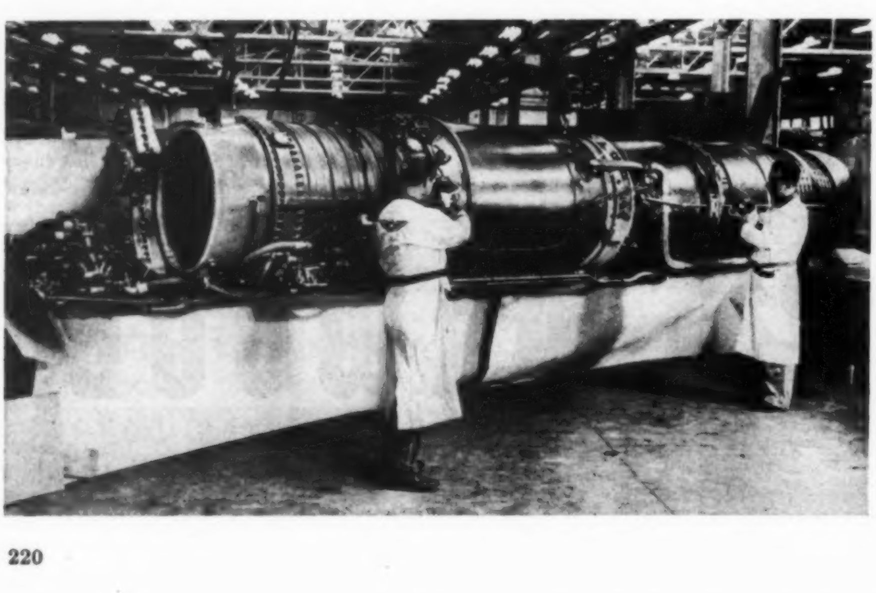
The engine is of axial flow or straight-through design. Approximately 25 feet long and 40 inches in diameter, it is extremely lightweight and produces more thrust per square inch of frontal area than previous turbojets. An afterburner, which reheats the exhaust gases after they leave the turbine but before they emerge as a jet stream, is employed to boost power.

Substantial amounts of columbium and cobalt have been elim inated in the new J40, and eve greater savings of these and othe scarce materials are to be effecte in subsequent versions of the en gine.

It is expected that several ne fighter planes scheduled for earl production for the Navy will b powered by this engine. Include are the McDonnell F3H Demon an the Douglas F4D Skyray.

Technical Manpower Shortage Discussed

One of the nation's top problems—the shortage of technical manpower—was the subject of the Engineering Manpower Commission conference held recently in Chicago. Appealing for a program to deal with this "alarming" situation, conference speakers cited the acute shortage of engineering and scientific personnel, which is expected to become worse in the next few years.



Because of the shortage of engi neering personnel, conferenc speakers said, industry is har pressed to maintain the flow of de fense and civilian materials, an development has been impede There is an existing deficit of en gineers estimated at 50,000 and a annual demand for at least 30,00 engineer graduates in industr alone. Of 30,000 engineers grad uated in 1952, 5000 have militar obligations to fulfill as a result o ROTC programs, and others ar being drafted into military service Next year's graduates will includ only about 20,000 engineers, and i 1954 it is estimated that there wi be a scant 17,000. Engineerin graduates will be even fewer i there are draft withdrawals befor graduation.

The gravity of the situation i apparent when it is remembere that military, industrial and gov ernment needs must be met by thi insufficient number of engineerin graduates. Interests of all thre of these groups were represente at the manpower conference, an

the EMC looks for a natoinal program of manpower utilization which will benefit "national health, safety and interest."

TV Camera for Nonbroadcast Use

Directed toward facilitating the adoption of closed-circuit television for industrial applications, the Tube department of RCA Victor has announced its first commercial version of the Vidicon industrial TV camera tube. Revealed as a developmental project approximately two years ago, the small electronic "seeing eye" is claimed to make possible compact, simple, lower cost television camera equipment for nonbroadcast use.

Size of Vidicon tube as compared to an image orthicon broadcast television camera tube

Somewhat similar in size and shape to a cigar, the tube is only one-tenth the size of a standard broadcast television camera tube. It is 1 inch in diameter and 6 inches long. The tube's small size and simplicity make possible the design of a television camera approximately the size of a 16mm home movie camera.

Such a camera can be placed to permit close-up or wide-range televising. It is expected to find application in production control and observation of potentially dangerous experiments.

Utilizing a photoconductive layer as its light-sensitive element, the camera has a sensitivity which permits the televising of scenes with 100 to 200 foot-candles of incident illumination. The photoconductive layer has a spectral response characteristic approaching that of the human eye. Operating with relatively low dc voltages, the tube provides 400-line resolution, employs magnetic focus and magnetic deflection.

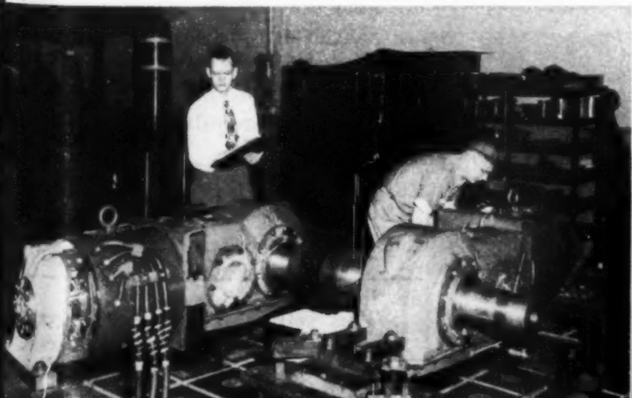
Install Motor Development Test Facilities

New test facilities for motor engineering development have been put into operation at the Locomotive and Car Equipment department of the General Electric Co. in Erie, Pa. Traction motors and generators for locomotives larger than now being built can be tested here.

The new facilities comprise eleven test tables, a soundproof room for noise tests, an air chamber for motor-ventilation tests and life-test facilities for simulating service conditions. A substation which covers 4500 sq ft of the 20,000 sq ft installation supplies power.

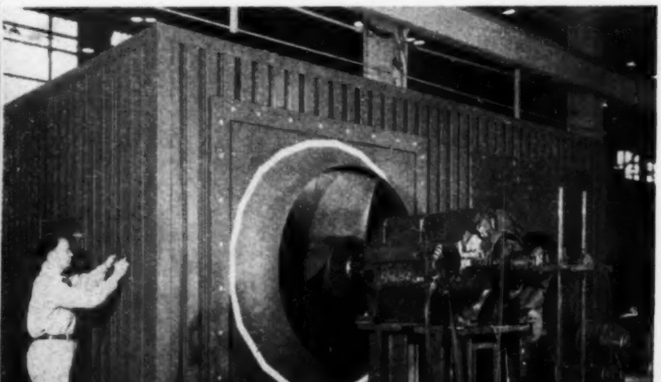
All eleven test tables were built with similar configurations of switches and controls to permit an interchange of operating personnel with a minimum of special instruction. Although each table was designed to handle a particular type of equipment, they are sufficiently flexible so that testing of other types may be done.

Sound level in the sound testing room has been reduced from 90 to 60 decibels so that sound will not affect test results. Walls of the room are insulated with rock wool, and the room floats on a two-foot bed of sand to isolate it from the building foundation. The roof is removable to permit use of an over-



Two G-E test engineers, above, conduct a load test on double-reduction geared motors

Plenum air chamber, below, in which fans and blowers of various types are tested



head crane.

A large plenum air chamber, partitioned into two 10-ft cubicles by a steel wall containing nine standard calibrated nozzles, is used to study air flow requirements for ventilating traction motors and generators and for studying the effectiveness of different fan and blower designs. Sizes of the spun aluminum nozzles range from 2 to 16 inches in diameter. Air flow can be measured accurately up to 30,000 cfm by opening and closing various combinations of nozzles.

There are large openings in the front wall of each cubicle for installation of fans and test motors, so that various conditions of intake and exhaust can be created. The difference in static pressure between the two chambers is measured by averaging pressures from 18 different locations in each chamber.

Life-test facilities permit simulating actual road conditions for a wide range of equipment over varied cycles of operation. Such tests are accelerated by the use of automatic control, which also permits study of fatigue and aging under operating conditions. Motors can be tested on an outside platform to determine the effect of normal weather changes.

Among the areas to be covered by air-flow tests are air flow versus speed of self-ventilated machines, and air flow versus static heads developed at constant speeds for blowers on force-ventilated equipment.

Speed-torque characteristics will be studied by measuring generated voltage at constant speed for prescribed loads and field excitations. Constant speed can be held to within 1/8-rpm accuracy.

Large Mill Rolls Aircraft Skin Sheets

Designed, installed and operated by the Aluminum Co. of America under Air Force contract, a new rolling mill will produce skin plates up to 10 feet wide and more than 33 feet long. Present limits are approximately 5½ by 25 feet.

This is the first mill to be built in this country specifically for the rolling of tapered aircraft skin sheets. It is to be installed at Alcoa's Davenport, Ia., plant and is expected to reduce assembly time for airplane wing sections.

Thickness of finished plates will range from 0.064 to 1.5 inches, and the maximum taper ratio will be 4.5 to 1, with a maximum tapered rate of 0.050-inch per foot of sheet.

Capable of a maximum delivery speed of 120 fpm, the 114-inch, four-high reversing mill will be driven by a General Electric 300 hp, dc motor. Speed of the moto ranges from 80 to as low as 5 rpm because mill speed must be main tained in ratio to the speed of mo tor-driven screws which adju roll separation to produce desire plate thickness.

Alpha-Particle Isotope Now Available

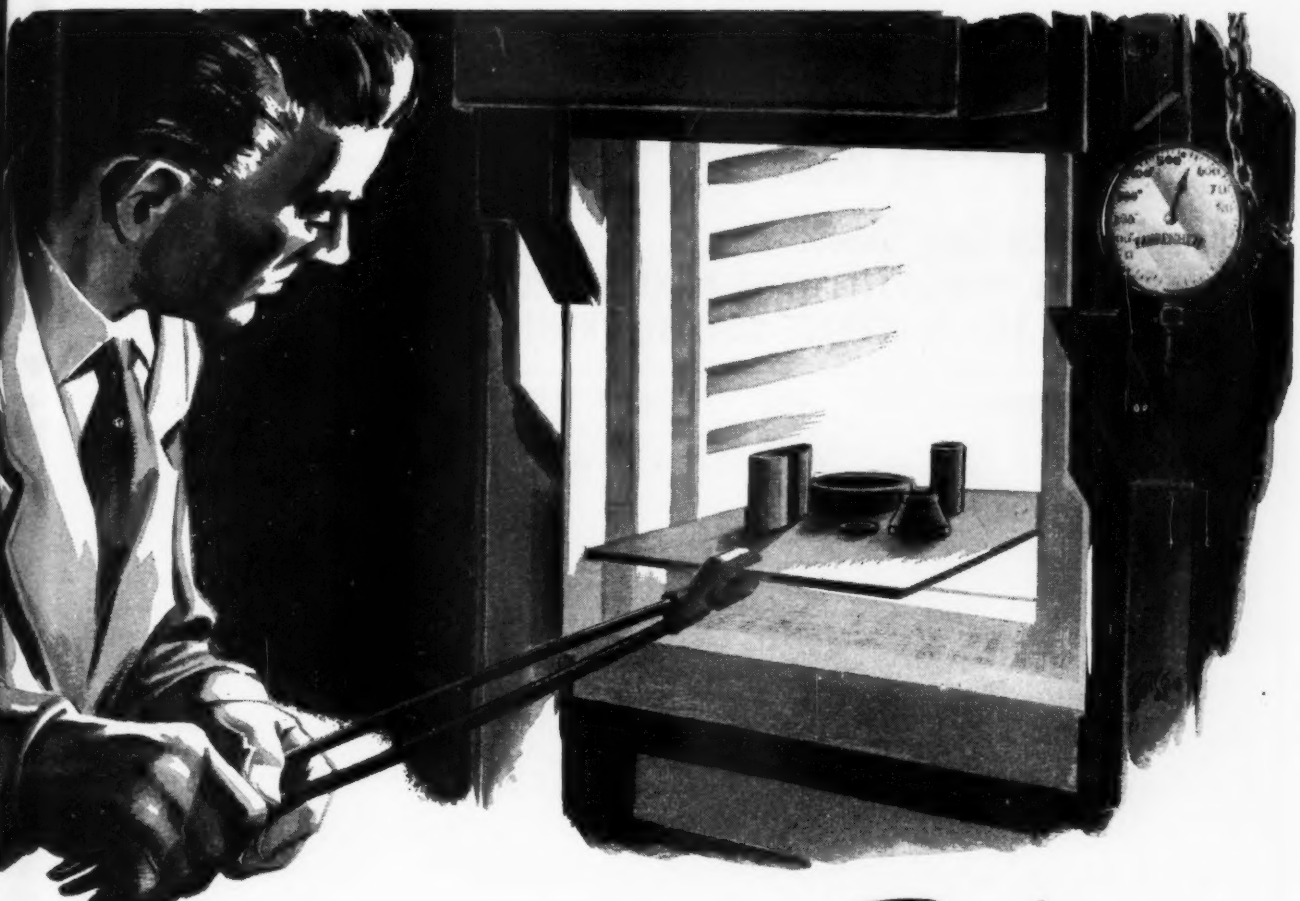
Reactor-produced radioactive po lonium-210 may now be purchase at Oak Ridge national laborator for research activities. This typ of polonium is the first reacto produced radio isotope to be sol which emits alpha particles. It a so can be used as a source of high energy neutrons.

Announcement of the availabilit of the new isotope marks the thir anniversary of the isotope pro gram, during which Oak Ridge the world's principal isotope pro duction center, shipped more tha 35,000 individual orders of isotope to users in 46 states and 33 for eign countries. The demand for iso topes has risen at a steady rat each year. No indication of leve ing off is apparent from year-to year figures on shipments. Ship ments for 1952 total some 9,000 about 25 per cent of all the ship ments in the entire six years o the program.

The new isotope can be used i luminous phosphors, static elimi nation devices, and for other indus trial purposes. Present day us however, is expected to be for re search in physical and biologica investigations, since the isotope i not at present available in th quantities required for more com mon uses.

Polonium-210 is available in tw forms. As a neutron source, th

This self-contained Hydropress oi hydraulic extrusion press is designe to provide stepless adjustment o extrusion speed from both the mai operating pulpit and from an auxi iary control stand at the plate exit. The press is to be used fo the extrusion of magnesium an magnesium alloys at the new Madi son plant of the Dow Chemica Corp.



THE MAGIC OF *Silicone*

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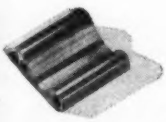

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SILICONE PARTS FABRICATED BY STALWART assure maximum performance and maintain their desirable physical, chemical and dielectric properties under critical operating conditions. In many applications, they have replaced metal fabrications forced upon manufacturers by the limited thermal stability of organic rubbers. Silicone rubber parts now are used for many difficult applications . . . aircraft seals and instruments, marine equipment, transformers, internal combustion engines, motors, generators, chemical and electronic equipment, etc.

EXPERIENCED STALWART ENGINEERS are prepared to mold, extrude, punch, machine-cut and mandrel-build precision silicone rubber parts to meet individual as well as S.A.E. and A.S.T.M. specifications.

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polonium will be mixed with beryllium and enclosed in a cylinder o nickel about 3/4-inch in diameter As an alpha particle source, th polonium will be placed on a stri of platinum.

High-Temperature Lubricants Extend Motor Service Range

Greater efficiency in the opera tion of electric motors may be on of the results of the current Navy Industry program devoted to th development of new high-temper ature lubricants. With the new high-temperature lubricants now commercially available, electric motors can be operated continu ously at 150 C (300 F).

New high-temperature materials such as silicone-glass insulation have permitted substantial in creases in power output without increase in physical dimensions by allowing electric motor and other rotating electrical equipment to op erate at higher temperatures. The development of such equipment however, was retarded by the lack of information on the operation of antifriction bearings and greases at elevated temperatures—150 C and higher. The new lubricants developed as a result of the joint program will allow operation of electric motors at these high tem peratures with intervals of 500 to 1000 hours between relubrication.

These intervals can be much longer if the motors are operated below the maximum temperature of 150 C for over 50 per cent of the time. Some motor design changes have been indicated by the study which may further extend the intervals between lubrication. Previous operating temperatures on a continuous basis were limited to about 80 C.

Nine newly developed greases— all of which are now commercially available—were investigated and tested. These included a mineral oil gelled with strontium soap, two mineral oils gelled with sodium soap, a mineral oil and diester blend gelled with lithium soap, and four silicone greases, one of which was gelled with carbon black and the others with lithium soap. The silicon-lithium soap greases orig-

inated by the Naval Research Laboratory were found to be useful at 150 C for over 500 to 1000 hours of operation without regreasing. In occasional tests good performance for as long as 6000 hours was observed.

Lowering the operating temperature to 125 C resulted in a large increase in the life of all the greases tested, the silicone-soap greases still having a clear advantage over all the others. Several nonsilicone greases, however, gave dependable operation at 125 C for 1000 to 2000 hours without regreasing. For operation at 100 C, the three soap-gelled silicone greases and three of the nonsilicone greases gave 10,000 to 15,000 hours of satisfactory operation without relubrication.

A military specification incorporating the findings of this cooperative program has been issued by the Navy Bureau of Ships. Another outcome of the study, however, has been the indication of necessary changes in motor manufacture to make best use of the new greases. For maximum utilization of Class H insulation in electric motors, the study indicated that the lubricant system should be redesigned so that more of the fluid bled from the grease migrates to the bearings. Seals or shields to prevent loss of grease from the bearings, a larger reserve of grease within the bearing proper, ball cages of selected nonferrous metals, and use of bearing steels stable at higher temperatures were the general factors deemed to be essential in future motor designs.

Outline Industrial Research Program

At the Centennial Conference on Industrial Research, sponsored by Armour Research Foundation, 350 research executives met recently to consider the question "Where do we go from here with industrial research?" General Electric vice president Harry A. Winne, one of the principal speakers, outlined a five-point program for industry, saying that industry must answer "yes" to the following questions:

1. Will industry continue to in

vest an adequate portion of its profits in research?

2. Will industry cultivate in its top management a full understanding of the essential characteristics of—and relationships between—fundamental research, applied research and advanced engineering development?

3. Will industry foster the kind of intellectual and physical environment that is essential to creative research?

4. Will industry make a place for the free and uncommitted investigator?

5. Will industry strengthen that vital link in the chain—a strong advanced engineering organization capable of translating research results into industrial practice?

A sound research and development program co-ordinated with business interests was termed as "a form of profit insurance and a continuing operating necessity."

Lincoln Announces Contest, Makes Awards

A new Mechanical Design Award program and results of the fifth Annual Engineering Undergraduate Award and Scholarship Design program have been announced by the James F. Lincoln Arc Welding Foundation of Cleveland.

The Mechanical Design Award program just announced is a competition for designers, engineers and manufacturers of machinery of all types and offers $30,000 in 101 cash awards, as well as national recognition, for the best papers describing the mechanical design and construction of any type of machine or machine component which is designed for arc welded steel fabrication.

Any machine or component whose performance or appearance has been improved or whose cost has been reduced through the use of arc welding may be described; any person who has participated in the designing, planning or manufacturing of the machinery may compete in the program by describing his work. Papers, which will be judged by a jury of prominent engineers, educators, de-

Engineering News

signers and fabricators, may describe machinery in any one of eighteen classifications. Entries will be accepted until July 27, 1953.

Awards in the recent undergraduate competition were made for the best designs of machinery and structures stressing the value of arc-welded construction. Engineering undergraduates in 23 different schools received the awards, which total $5000, and scholarship funds totalling $1750 were awarded to the schools of the first three winners in their honor.

First award went to Richard Cheverton and Thomas Musslewhite of Georgia Institute of Technology for their joint design of a sub-calibre mount for a recoilless rifle. The design permits mounting a .22 calibre barrel in a 75 mm rifle for training purposes.

Merle Geoffrion, an undergraduate in the civil engineering department of the University of Delaware, received the second award for his design of a highway bridge.

The third award was made to Robert Murray, a student in the department of architecture of Rensselaer Polytechnic Institute. Mr. Murray made an analysis to determine the economical span length of standard sections.

The program is sponsored annually to encourage engineering students to study the possibilities of arc welding as an engineering tool for improving the quality of machinery and structures of all kinds while reducing costs. The sixth annual competition, for which entries are now being accepted, closes on June 29, 1953. All registered undergraduates are eligible to compete. Lincoln Foundation secretary A. F. Davis states that in order to encourage participation by all engineering undergraduates, the rules of the program have been simplified considerably to avoid conflict with regular school work and the restrictions of time.

Rules and conditions brochures for the Mechanical Design Award program, and rules booklets for the Engineering Undergraduate Award and Scholarship Design program are available from the Lincoln Foundation, Cleveland 17, O.

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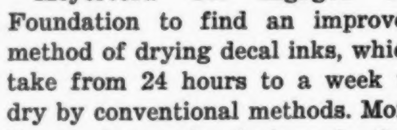
Engineering News

New Process Developed For Industrial Drying

Using a new process developed at Armour Research Foundation, paints, inks and varnishes can be dried in from 2 to 20 seconds. Since industrial drying methods employing heat or solvents require hours of drying time, this new process has the advantages of speeding production and releasing plant space formerly used for drying.

Sulphur-dichloride vapor used in the process reacts with pigment oils as coated materials are transported through an enclosed drying chamber on an endless belt. Drying time depends upon the paint, ink, or varnish used and the material being coated—paper, cloth, metal, wood, glass or plastic. After treatment in the chamber, coated materials are dust and tack-free and can be handled, stacked, packaged or given a second coat.

Meyercord Co. engaged the Foundation to find an improved method of drying decal inks, which take from 24 hours to a week to dry by conventional methods. More than a dozen chemicals and other methods of drying, including electron bombardment, infra red, ultra violet and dielectric heating, were investigated. Sulphur-dichloride vapor, which dries decal sheets in two



"Employment office? I'd like to hire an engineer with an inferiority complex. I won't be able to pay him what he's worth."

seconds, solved the problem, and the process was subsequently extended to other coatings and materials.

New Exhibits Added At Power Show

All branches of industry will be in the scope of the 20th National Exposition of Power and Mechanical Engineering, to be held in Grand Central Palace, New York, December 1 to 6, under the auspices of the American Society of Mechanical Engineers. To be held concurrently with the Annual Meeting of ASME, the show will have many exhibits exemplifying subjects under discussion.

The exposition covers all branches of the field of power from the conversion of energy into fluid forms to its application in moving mechanisms. Included will be the conversion processes themselves, distribution in transmission lines and pipelines, and many applications within the industrial plant.

Major exhibit classifications include power plant equipment, power transmission equipment, electrical equipment, heating, ventilating and air conditioning equipment, materials handling equipment, safety, building construction, plant maintenance, and equipment for research and testing. The classified list of exhibits comprises about 350 items.

♦

A free consulting service is being offered by **Bausch & Lomb Optical Co.**, Rochester, N. Y., in connection with its new industrial Stereomicroscope. Company representatives will demonstrate how various models of the new instrument can be built into machine, assembly and inspection setups.

♦

The **Society of Naval Architects and Marine Engineers** has announced the formation of a new regional section to be known as the Eastern Canadian Section. Headquarters will be in Montreal, and Richard Lowery, vice president of Canada Steamship Lines, has been elected chairman of the new section for the 1952-1953 season.

Engineering News

Huge Aircraft Test Structure Being Built

Designed for "destructive testing," the largest aircraft static-test structure outside of Wright Patterson Air Force Base is being built at the Douglas Aircraft Company's Long Beach division. When completed, the $200,000 installation, containing $50,000 worth of hydraulic jacks and electronic gages, will be able to tear a C-124 to pieces.

With girders 4 feet long and 2 feet wide, the static-test machine can maintain severe rigidity while sensitive instruments register the yield of a member under stress. The machine's concrete foundation is 10 feet deep and, with its embedded steel reinforcing, weighs 3000 tons. In addition there are 40 concrete piles extending to a depth of 20 feet underground which are capable of resisting a force of 50,000 pounds.

A steel track will be mounted on the foundation with 4500 drilled holes for the attachment by steel bolts of various hydraulic jacks and other testing equipment. These jacks can exert a force of from 10,000 to 100,000 pounds in any direction. Some are powered with electric motors; others are small enough to be operated by hand.

Rising nearly 40 feet toward the roof of the factory building, the overhead structure of the testing machine provides a clear span of 32 feet.

Indicates Business Has Tripled Since 1930's

The tremendous increase in business during the past dozen or so years is indicated by a recent announcement from the American Gear Manufacturers Association. Future base period for the Association's index will be the calendar years 1947, 1948 and 1949. Reason for the change is evident when the index figure for July, 1952, is considered. Based on the new figure equalling 100, July's index is 145.3; based on the old 1935-1939 period the same index would be 453.9.

Engineering News

Westinghouse To Develop Large Atomic Engine

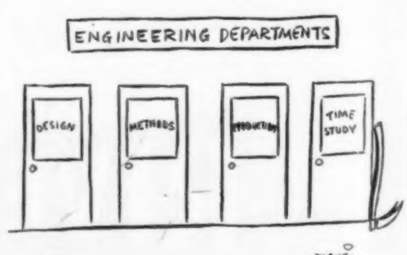
An atomic power plant "suitable for the propulsion of large naval vessels such as aircraft carriers" will be built by Westinghouse Electric Corp. The present contract between the Atomic Energy Commission and Westinghouse has been modified to include development work on such a nuclear reactor, it was recently announced by Charles H. Weaver, manager of Westinghouse Atomic Division. Work on the new reactor project will be centered at the Atomic Power Division Bettis plant near Pittsburgh.

♦

At the 24th Annual Meeting of the **Gray Iron Founders' Society,** held October 16 and 17, the following officers were elected for 1953: president, H. J. Trenkamp, Ohio Foundry Co.; vice president, T. I. Curtin, Waltham Foundry Co.; secretary, C. H. Ker, Dalton Foundries Inc.; and treasurer, W. O. Larson, The W. O. Larson Foundry Co.

♦

Shakeproof Inc., a division of **Illinois Tool Works,** Chicago, has announced the opening of a new plant in Des Plaines, Ill., for the production of metal fastening devices and special precision stampings. This new plant makes possible an expansion of the defense tooling program of the parent company. Manufacturing space made available in the Chicago plant will be devoted to the production of gear measuring machines, metal cutting tools and machine tools.



Engineering News

New Jet Plane Completes Carrier Tests

With more than 370,000 engineering hours behind it, the swept-wing XFJ-2 Fury has successfully completed its first carrier suitability tests. This airplane, which is in the early stages of production by North American Aviation, is an advanced version of the FJ-1, the Navy's first operational jet. Completely redesigned, it incorporates many improvements, including an especially designed wing with a 35-degree sweepback and a hydraulically powered "Flying Tail."

•

The saw manufacturing concern of E. C. Atkins and Co. of Indianapolis was recently acquired by **Borg-Warner Corp.,** Chicago. It will be operated as the Atkins Div. of the parent company.

Report on Materials

Ending the Controlled Materials Plan is going to be a touch-and-go affair. As late as the middle of October, most indications pointed toward a termination of controls on the major metals by mid-year, 1953. But later trends seem to reverse this point of view.

Here is the situation. After the setbacks in steel production caused by the steel strike, recovery was much more rapid than expected by both industry and government forecasters. Steel production quickly rose to a record rate, and is expected to continue to a new high of 116 million tons by the beginning of next year.

This fact, coupled with estimates by NPA that manufacturers would not be able to use up uncashed third and fourth-quarter CMP tickets before the first quarter of 1953, led to a rosy picture of the steel supply situation. Accordingly, definite plans were made to relax some controls on construction by May 1; automobile manufacturers were cut to a 60 per cent first-quarter allotment, since third and fourth-quarter allotments were expected to carry production into next year; and general feeling in

Engineering News

Washington had NPA getting ready to pack its trunks and silently steal away.

Now, there is a serious doubt whether all is calm and peaceful. Automobile manufacturers have used up their third and fourth-quarter allotments much faster than forecasted; latest returns indicate that these tickets will all be used in the fourth quarter, and that the 60 per cent allotment of steel by NPA will not be enough the first quarter. There is some suspicion that the same situation will be true in other industries.

So it seems that decontrol estimates will again be tossed back into the laps of the crystal-ball gazers. Possibly the pessimistic estimates calling for decontrol in late 1953 or 1954 which were current at the end of the steel strike may not be so wrong after all. If, however, manufacturers are forced to stay within their current allotments—probably a highly unpopular method with industry—the situation might rectify itself faster.

Copper Still Tight: Although recommending continuance of studies for "eventual" decontrol, NPA's advisory committees from industry are not optimistic on the supply situation in copper. Thus the Wire and Cable committee recommends no change in the present method of allocating copper, pointing out that copper is not in plentiful supply now and probably won't be until some time in 1953. The Brass Mill Industry committee apparently agrees and also asks for no change in allocating procedure.

Primary copper producers, however, blame government activities for limiting the supply. Price of copper seems to be the biggest deterrent. "We are now in the stage where, if controls don't work, the thinking is that new controls should be added," said several industry advisory committee members. "Copper prices should be decontrolled and it is absolutely necesesary to get rid of the 24½ cents per pound price in order to reach the production goals in refined copper." In addition to price decontrol, copper producers feel that stockpile purchases should be deferred until the price situation has

been resolved.

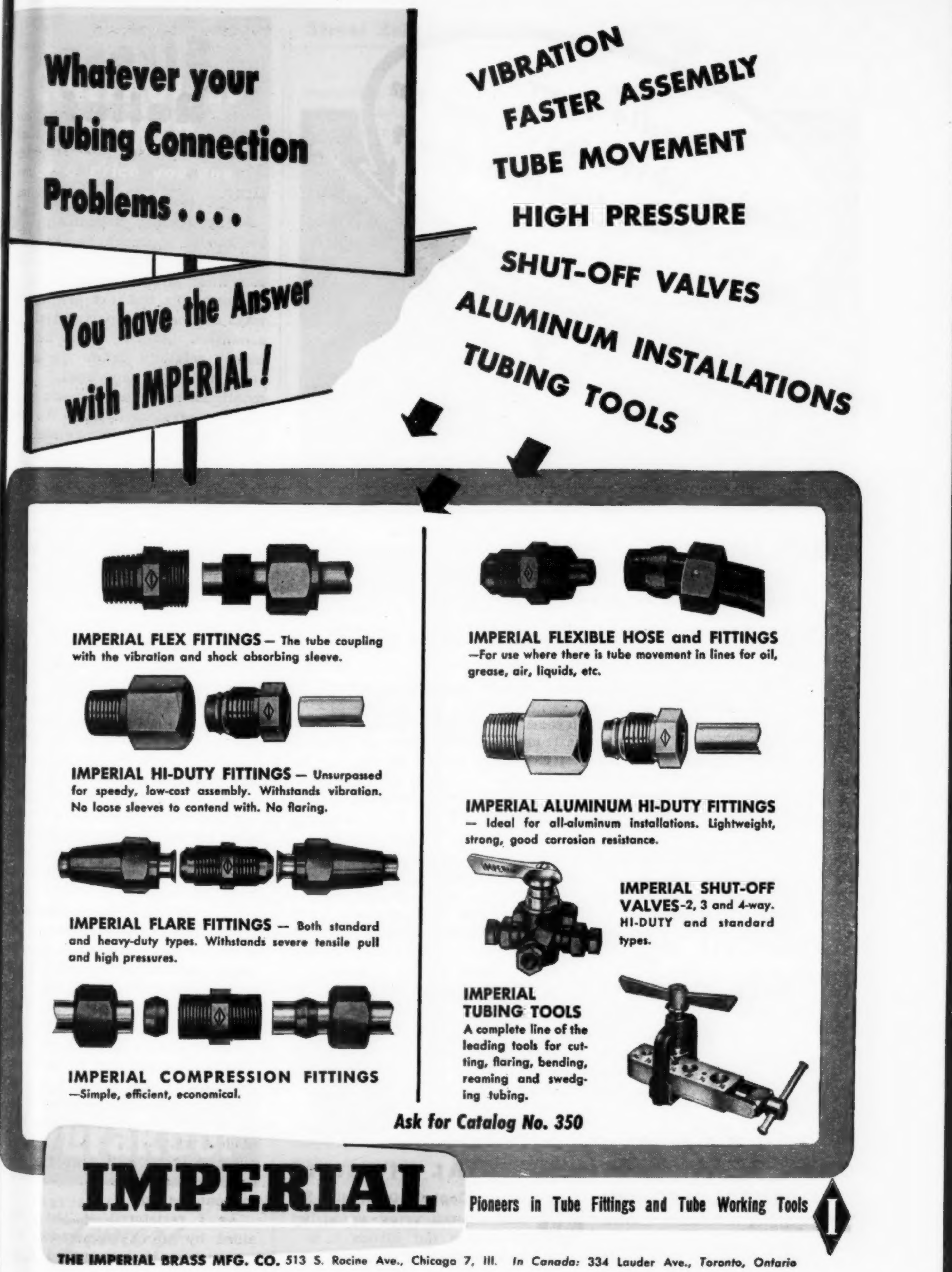
Zinc Supplies Ample: Many engineers apparently still feel that the zinc supply situation is tight, although such is not the case. Government restrictions on the use of zinc (allocation, delivery and use controls) ended last May, because of an improved production and supply situation. According to the American Zinc Institute, this improvement was not temporary, and for the foreseeable future zinc reserves are adequate for all needs.

The Institute points to these facts: (1) Zinc is no longer classified as a critical and scarce material; (2) The government is no longer stockpiling zinc regularly; (3) Slab zinc production is high, stocks are adequate, and price is low; (4) World zinc mine reserves are adequate for expected future needs. Thus, the Institute says, "conservation" of zinc is no longer necessary.

In a Pickle Over Nickel: Any apparent improvement in nickel supplies, NPA recently told members of the Stainless Steel advisory committee, is the result only of temporary adjustments and changes in certain demands and programs. Stockpile requirements for nickel are still as great as ever, and military and Atomic Energy Commission requirements are likely to increase greatly. Demand for nickel for jet engines alone is expected to double by the fourth quarter of 1953.

Aluminum Supplies Reduced: Shortage of rainfall in the Pacific Northwest, which was expected at press date to cause a loss of somewhat under 90 million pounds of the metal, may in the next month cause an additional 30-million pound loss. Serious power shortages have occurred due to the lack of rainfall, greatly affecting aluminum production.

Last year, $4\frac{1}{2}$ million pounds of the metal were lost. This year, if the total loss reaches the 120-million pound figure expected by many, the total amount available will be cut some 5 per cent. Since aluminum demand has been on the upturn in recent months, this shortage could have serious effects.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Stress Relief

THE invincible logic of J. P. Henderson's arguments and his candid approach to the facts of everyday living are usually somewhat overwhelming. As in the past, his remarks are not always confined to departmental personalities. At times, the field of human relations draws his attention—and his appraisal. This month he recommends some practices in personal living. We suspect the issue is controversial.

"Front, Boy!"

Several weeks ago, I had to take a trip to New York. In the club car going down, I sat among a group of young men who, it turned out, were chemists. Along with more general conversation they discussed the difficulties of ever being anything but salaried employees. What chance did they have of going into business for themselves? What could they do but work in industrial laboratories or research organizations, or teach?

Setting up in business these days was too difficult, too complicated. It not only took a lot of capital, but too much specialized knowledge. Before they could buy a test tube or hire a flunky, they should employ a tax expert, a personnel man with knowledge of labor laws and several expert accountants.

One chap, presumably in his early thirties, expressed the desire to gamble just once in his profession before settling down to the struggle of climbing the hill of the salaried employee. Wasn't there some way he could take a long chance in advancing himself professionally, gambling on making more money legitimately? If it failed—well, he would at least have made a try.

I left them, still arguing, with their strange chemical concoctions in front of them.

As I registered at the hotel, aided by an ex-ambassador with a carnation, he commanded in well

modulated tones, "Front, boy."

Right then I knew I had the partial answer to the chemist's or other technical men's problem. It was the succinct answer I had been looking for—*Front*.

How can a young employee take a gamble that might pay off in more rapid promotion? *Front* is the answer, and here is what I mean by it.

A young man right out of college takes a job and works up in a modest way to a certain position. Sooner or later, he may change jobs, moving to a new location.

While it is not expedient for everyone to move around, if such a change is made, it has some advantages that often are not exploited. One advantage of a move rests in leaving all of one's early mistakes behind. His new associates never saw him as a youth.

Is he a big shot, or just average stuff? Sooner or later, the new employee is going to be classified by his bosses. What is easier than for them to take him simply at his own valuation?

The point is, we are all mentally lazy. We don't stop to analyze and classify people accurately; we just label them unconsciously. Assuming a young man has the stuff, if he acts like a reasonably big shot, we think he is a big shot.

So, he moves to the new job and rents or buys a house. What he should realize is that if he chooses to live in a dump, he is likely to harm his prospects. Starting out just fresh from college, he lived modestly; with high expenses he skimped, accumulated some possessions and saved a little money. The move is a fresh start. Can he barely afford a *good* house, in the right neighborhood? Will they be proud to entertain there? Are there some modest clubs to which they could belong?

There's where the gamble comes in; he's risking his financial picture in the hope that he can establish himself on a certain level and make a reasonable impression as an up and coming young man. It's *front,* and because we are all mentally lazy, he might convince us.

If I should tell my next-door neighbor who is in the insurance

Stress Relief

business, that I was handing out advice like this, I would be thoroughly scolded. "That's what's wrong with this country today," he would say, "Too many people trying to live beyond their income, putting on front instead of money in the bank."

That's probably true in general, and many people need this opposite advice. But the technical man seems to lean over backward in his conservative attitude towards front. What the technical man must realize is this:

Before you can occupy an executive position, your boss must be able to *imagine* you in such a job. He must have a mental picture of you behind a mahogany desk, receiving important visitors graciously.

Riding to work in your sweat shirt on that old bicycle, from your hovel down by the city dump, makes it very difficult for the president to consider moving you into that thickly carpeted vacant office next door to him. He may have a lot of imagination, but not that much.

—J. P. HENDERSON

They Say . . .

"You [the engineer] therefore owe everything that you have to give in honest and sincere effort toward engineering progress. Dishonesty is inconceivable in even the smallest detail of a professional mission. Carelessness and sloth are but little less reproachable. Integrity and energy deepen the luster placed upon the profession of engineer by the master builders of the past. Disclosure of advances must be given freely to aid in educating the engineers of the future. Thus with each generation rising above the teaching of its forerunner, progress by employment of science is the gift of engineering to mankind. With this gift must go the engineer's unending effort that ethics may mark its use and morality remain its master."—L. E. GRINTER, *vice president, American Society for Engineering Education.*

Meetings

AND EXPOSITIONS

Nov. 10-14—

National Electrical Manufacturers Association. Annual meeting to be held at Haddon Hall, Atlantic City, N. J. W. J. Donald, 155 East 44th St., New York 17, N. Y., is secretary.

Nov. 12-15—

Society of Naval Architects and Marine Engineers. Sixtieth anniversary meeting to be held at the Waldorf Astoria Hotel in New York, N. Y. Additional information may be obtained from society headquarters, 29 West 39th St., New York, N. Y.

Nov. 19—

American Standards Association. Annual meeting to be held at the Waldorf Astoria Hotel, New York, N. Y. Additional information may be obtained from society headquarters, Grand Central Terminal Office Bldg., 70 East 45th St., New York 17, N. Y.

Nov. 20-21—

American Society for Quality Control. Seventh midwest conference to be held at the Claypool Hotel, Indianapolis, Ind. Additional information may be obtained from society headquarters, 70 East 45th St., New York 17, N. Y.

Nov. 29-Dec. 4—

American Society of Mechanical Engineers. Annual meeting to be held at the Statler Hotel, New York, N. Y. C. E. Davies, 29 West 39th St., New York, N. Y., is secretary.

Dec. 3-5—

American Management Association conference to be held at Hotel Statler, Cleveland, O. Additional information may be obtained from society headquarters, 330 West 42nd Street, New York, N. Y.

Dec. 3-5—

Society for Experimental Stress Analysis. Annual meeting and ex-

Meetings and Expositions

position to be held at Hotel McAlpin, New York, N. Y. W. M. Murray, Central Square Station, P. O. Box 168, Cambridge 39, Mass., is secretary-treasurer.

Dec. 4-6—

American Institute of Mining & Metallurgical Engineers. Electric furnace steel conference to be held at the William Penn Hotel, Pittsburgh, Pa. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18, N. Y.

Dec. 15-17—

American Society of Agricultural Engineers. Winter meeting to be held at the Edgewater Beach Hotel, Chicago, Ill. Raymond Olney, P. O. Box 229, St. Joseph, Mich., is secretary.

Jan. 12-16—

Society of Automotive Engineers. Annual meeting and engineering display to be held at the Sheraton-Cadillac Hotel in Detroit, Mich. Additional information may be obtained from society headquarters, 29 West 39th St., New York 18, N. Y.

Jan. 19-22—

Plant Maintenance Conference. Fourth annual conference to be held at the Public Auditorium in Cleveland, Ohio. Additional information may be obtained from the exposition management, Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

Jan. 21-23—

Society of Plastics Engineers. Ninth annual technical conference to be held at the Hotel Statler in Boston, Mass. Additional information may be obtained from society headquarters, 409 Security Bank Bldg., Athens, Ohio.

Jan. 26-30—

International Heating and Ventilating Exposition. Eleventh exposition to be held at the International Amphitheatre in Chicago, Ill. Charles F. Roth, 480 Lexington Ave., New York 17, N. Y., is manager.

Design Abstracts

(Continued from page 174)

capacitor. An electrical circuit analogous to the original motor can easily be established as,

$$e_t = \left[e + T_m \frac{d}{dt}\left(e + T_a \frac{de}{dt} \right) \right] + R_a \left(i_L + T_a \frac{di_L}{dt} \right)$$

where lower-case symbols indicate corresponding analogs. Comparing the equation for this analog with that derived for the motor it may be noted that they match, term by term: $e_t \equiv E_t$, $e \equiv E \equiv g\omega$, and $i_L \equiv M_L/b$. Also, terms for the mechanical time constant, T_m, and the armature time constant, T_a, have been incorporated. Thus, the analog is a suitable representation of the motor.

A similar procedure can be followed for the generator. Here a field voltage, E_f, produces an internal generator voltage, E_g. Load current through the armature resistance and inductance reduces this voltage to the terminal voltage, E_t. Also, induced and field voltages can be related to the field current. Equations for induced voltage as a function of field current; and for the effect of armature voltage drop may be written as

$$E_f = \frac{R_f}{K} \left[E_g + \frac{L_f}{R_f} \left(\frac{dE_g}{dt} \right) \right]$$

$$E_t = E_g - R_a \left[I_a + \frac{L_a}{R_a} \left(\frac{dI_a}{dt} \right) \right]$$

These are the generator equations.

To represent a portion of the system gain, an amplifier is incorporated in the generator analog and the terms written as:

$$e_g = -Ae_o$$

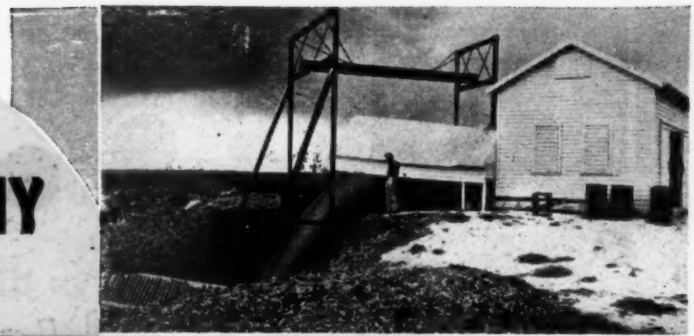
$$i_1 = \frac{1}{R_1}(e_f - e_o)$$

$$i_2 = \frac{1}{R_2}(e_g - e_o)$$

$$i_3 = C\left(\frac{de_g}{dt} - \frac{de_o}{dt} \right)$$

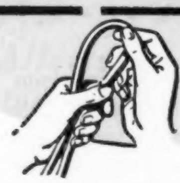
$$i_1 + i_2 + i_3 = 0$$

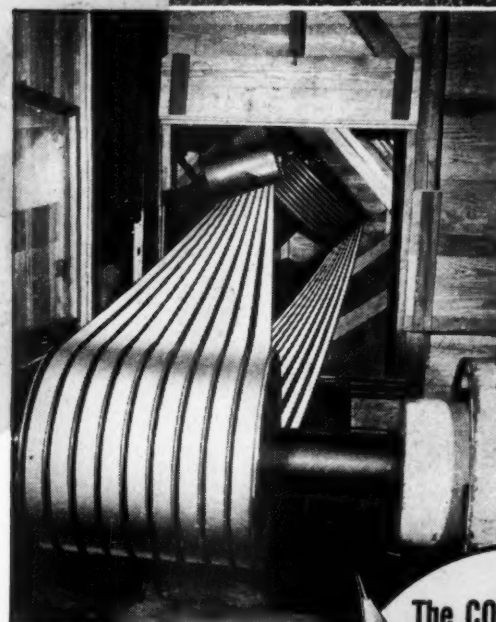
Input and output voltages of the



See for Yourself WHY

Gates Vulco Ropes Wear Longer





This big 80,000 GPM turbine pump lifts water against a 14 foot head from a drainage canal to Lake Hicpochee in the Diston Island Drainage District near Moore Haven, Fla. Driven by a diesel engine through a 9 D-Section Gates Vulco Rope Drive with a one-eighth turn, the pump drains 19,000 acres of land for sugar cane and cattle raising.

Make this Simple Test

Take any V-belt that has *straight* sides. Bend that V-belt while you grip its sides between your fingers and your thumb. You will *feel* the sides of the belt *bulge out*—as shown in figure 1-A, below.

Clearly, that outbulge forces the sides of the belt to press *unevenly* against the V-pulley —and this concentrates the *wear* where the bulge is greatest.

Now, make the same test with the belt that is built with the Concave Sides—the Gates Vulco Rope.

Figures 2 and 2-A show clearly what happens when you bend a Vulco Rope. Instead of *bulging,* the precisely engineered Concave Sides merely *fill out* and become perfectly straight. This belt, when bent, precisely fits its sheave groove.

Because there is *no bulging,* the sides of the Gates Vulco Rope always grip the full face of the V-pulley *evenly* and therefore wear *uniformly*—resulting in *longer belt life* and *lower belt costs for you.*

When you buy V-belts, be sure to get the V-belt with the Concave Sides--the Gates Vulco Rope!

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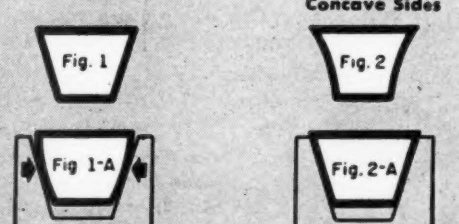
What Happens When a V-Belt Bends

Straight-Sided V-Belt | **Gates Vulco Rope with Concave Sides**



How Straight-Sided V-Belt Bulges in Sheave-Groove. Sides Press Unevenly Against V-Pulley Causing Extra Wear At Point Shown by Arrows.

The Concave Sides Fill Out to a Precise Fit in the Sheave-Groove. No Side Bulge! Sides Press Evenly Against the V-Pulley — Uniform Wear — Longer Life!

CS-528-A

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generator analog are given by

$$e_f = -\left(\frac{R_1}{R_2}\right)\left[e_g + R_2 C\left(\frac{de_g}{dt}\right)\right] - \frac{1}{A}\left(1 + \frac{R_1}{R_2}\right) \times \left[e_g + \frac{R_2^2 C}{R_1 + R_2}\left(\frac{de_g}{dt}\right)\right]$$

Because the amplifier gain, A, is very large (about 5000) the second term of the equation is unimportant and can be neglected. Thus

$$e_f = -\left(\frac{R_1}{R_2}\right)\left[e_g + R_2 C\left(\frac{de_g}{dt}\right)\right]$$

Also

$$e_t = e_g - R_a\left[i_a + \frac{L_a}{R_a}\left(\frac{di_a}{dt}\right)\right]$$

A comparison of the generator equations with these analog expressions shows that the analog is suitable for the generator.

To construct the analog for the entire regulator system, the analogs of the various components are assembled as shown in *Fig.* 3. The relationship between the analog and the regulating system is the same as exists between the ac network calculator and the system it represents—it is an easily reconstructed miniature of the system.

With this analog, different operating conditions can easily be simulated on the analog computers. For example, the effect of applying load can be obtained by closing a switch (*Sw* 1) on a high resistance in series with a large voltage. Opening another switch (*Sw* 2) simulates a sudden change of reference. The regulating system can be subjected to conditions of disturbance akin to those found in actual operating conditions and the response of the regulator can be noted. The response of the tandem mill drive system, *Fig.* 1, to a suddenly applied change in voltage setting is shown in *Fig.* 4. Similar reactions of the mill to other changes in operating conditions are also easily simulated.

To have value, the computer's results must be translated into quality of end product by engineers intimately familiar with application and manufacture. Expe-

rience and common sense are the required factors.

Analog computers have played a vital role in the development of regulating systems by reducing the major obstacle in the analytical process. The ability of the computer to evaluate system performance rapidly and economically allows the designer more flexibility and an opportunity to try out systems and ideas too radical or novel to build on a full scale without confirmation.

From an article entitled "The Role of the Analog Computer in Designing Industrial Regulators" appearing in the Westinghouse Engineer for November 1952.

Dirty Oil Affects Bearing Performance

By H. Grady Rylander

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Department of Mechanical Engineering
University of Texas
Austin, Texas

SLEEVE bearings operated under normal conditions are always subjected to some foreign solid particles in the oil supply. Internal combustion engines produce solids in the combustion process, other particles are produced by wear, and others enter as dirt, rust and scale from the air or piping. A thorough understanding of the operating characteristics of sleeve bearings when subjected to certain particles in the oil leads to an improved design of bearings, shafts, oil, and oil filters.

The problem of filtering out small particles below 10 microns in size is extremely difficult, and in the majority of applications only a small per cent of the oil pump discharge is filtered. Even full-flow filters have by-pass arrangements to insure an oil supply to the bearings in the event the filter becomes clogged. Commercial automotive cartridge filters will not effectively remove particles below 8 microns and some will pass particles as large as 25 microns.

In most bearing test programs, the bearings are carefully tested under conditions such that ex-

tremely small amounts of dirt and impurities are passed into the bearing by the oil supply. This practice leads to design loads that are too high. It is now recognized by those who develop their designs under actual service conditions that such factors as fatigue strength, temperature limitations, friction, wear resistance, elasticity, anti-seize properties, and embedability are also important as well as load-carrying capacity. No one bearing material has been found that will solve all of these requirements.

Several of the bearing design conditions mentioned may be seriously altered when the oil is contaminated with solid particles. Bearing operating temperatures may be changed, wear increased, friction varied, embedability destroyed, and anti-seize properties changed by adding solid particles to the oil.

Test Methods: A test machine was designed to accurately measure torque produced by a single bearing. Steel-backed babbitt bearing inserts were used as test bearings and support bearings in the test machine. These inserts were 2.168-inches inside diameter by 1-inch long with a lead-base babbitt thickness of 0.020-inches. Loading was applied hydraulically through a linkage which provided a two to one increase in the load and was self-centering on the test shaft. Loads over 8000 pounds could be applied, although only 5000 pounds was used in the first series of tests due to limits on the bearing capacity.

For each particle type a new bearing was installed in the test housing and an unused portion of the test shaft was matched with the new bearing. Shaft rotation was started when the oil inlet temperature was within 4 degrees of 140 F. Then the test shaft speed was set, test load applied, and the oil flow regulated to approximately 3/4-pound per minute.

Data were recorded for varying speeds and loads to span values of ZN/P from about 5 to 100. In some cases the low values could not be obtained because of insuffi-

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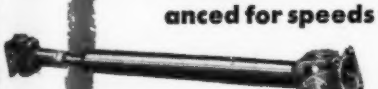

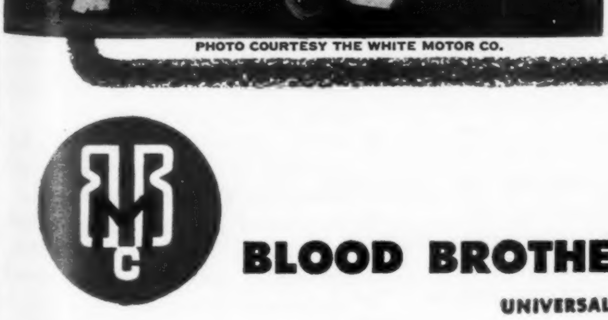

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cient power in the test drive, but these points were on the steep portion of the curve where marginal lubrication occurs. The experimental test program was arranged to obtain first a basis of comparison by using clean oil, then to proceed to soft small particles and increase the particle size, concentration, and hardness until some definite influence upon the bearing's operating characteristics was obtained. In order to carry out this program, particles and concentrations listed in TABLE 1 were tested in the order shown.

Table 1—Test Particle Sizes and Concentrations

Particle Material	Size (microns)	Concentration (gm per gal)
Graphite	2.5	0.47–1.87
Graphite	10	0.47
Graphite	30	0.94
Molybdenum sulfide	39	0.47–1.87
Red Rouge	1	0.47–17.0
Corundum	8	0.47
Corundum	11	0.47
Corundum	22	0.47

Test Results: The results of this investigation were dependent upon the accuracy of instrumentation and the care and procedure involved in conducting the test program. In order to improve the test accuracy, a large number of test points were recorded from which the following conclusions, based upon average operating conditions were obtained:

1. Soft particles of graphite and molybdenum sulfide in the oil supply to lead-base babbitt sleeve bearings will produce no improvement in the bearing operation.
2. Graphite particle size has no effect upon bearing operating characteristics when used in small concentration as they were in this program.
3. Graphite particle concentration will produce a change in the bearing oil flow characteristics. Increased concentration will increase the oil pressure required to maintain the same oil flow.
4. Coefficient of friction may be increased by increasing particle concentration.
5. Only the tests with small amounts of rouge caused friction values lower than clean oil.
6. Particles of hard substances must be larger than the minimum oil-film thickness to cause appreciable increases in wear or friction.
7. Wear may take place without a large increase in the bearing temperature.
8. Shaft wear may be much more rapid than bearing wear due to the babbitt embedding the large sharp particles.
9. Oil may be discolored by small concentrations of finely divided particles, but discoloration does not indicate the friction to be expected in a bearing. Very small amounts of hard particles will cause rapid wear even though the oil looks clean.
10. Filtration that removes any size particle will be helpful, but the removal of all particles larger than the minimum oil film thickness is required to stop wear and prevent increased friction. This means that for most high-load bearing applications, filtration must remove particles considerably smaller than 25 microns.

From a paper entitled "The Effects of Solid Inclusions in the Oil Supply to Sleeve Bearings" presented at the Semiannual Meeting of the ASME in Cincinnati, O., April 1952.

Hot Pressing Powder Metals

By Jerome F. Kuzmick
Consultant and President
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Clifton, N. J.

SIMULTANEOUS application of pressure and heat to metal powders—hot pressing—has for a number of years attracted the attention of powder metallurgists. Early experiments demonstrated the following possibilities for the hot-press method:

1. Pressure required for hot molding metal powders is considerably lower than is normally used for cold pressing.
2. Metal powders can be hot pressed to solid density in one operation without sintering.
3. Properties of hot-pressed alloys are similar to those of hot-forged or cold-worked annealed materials.

On the other hand, the early ex-

periments also pointed out some of the difficulties involved, such as:

1. Suitable die material to withstand the elevated temperatures, which generally are about the same as the sintering temperature.
2. Rapid heating of the metal powder and protection against oxidation.
3. Prevention of welding of the compact to the die as a result of hot pressing.
4. Rapid enough cycle to make the process practical.

Up to the present time, to the best of the author's knowledge, no process has been worked out and put into operation for rapidly producing hot-pressed metal-powder mechanical parts on a commercial scale. On the other hand, hot pressing has been successfully applied to a number of applications involving costly materials wherein this method is the only one which gives the desired results and cost is secondary.

It is known that considerable experimental work is being done on extending the applications of hot pressing. Generally the method follows one of two types:

1. Hot forging. In this method normally cold-pressed and sintered parts are removed hot from the sintering furnace and subjected to a forging operation, usually in water-cooled steel dies.
2. Hot molding. In this method, pressing and sintering are simultaneously performed in one operation. The charge in the die may be the metal powder itself, or it may be a preform which has previously been molded by the usual cold-pressing method.

Types of Hot Presses: The simplest type of hot press consists of an ordinary vertical electric furnace built into a hydraulic or otherwise actuated press. Such apparatus has often been used for experimental and semiproduction work; however, in most cases, the rate of heating is too slow and the setup is too unwieldy to permit economical production.

Another type often used in this country and also abroad is one in

Design Abstracts

which a carbon mold is heated by its own resistance to the passage of an electric current. These resistance type hot presses are very useful for fairly rapid hot pressing of relatively small sections. Carbide sections up to 4-in. diameter and 4-in. tall have been used. The heating rate is very rapid; small sections can be heated from room temperature to 2500 or 2600 F in less than 1 minute. By varying the mold cross-section, it is possible to concentrate the heat in any desired section. This technique is useful in producing composite materials in which one section may be made of an alloy requiring higher pressing temperatures than the rest of the piece.

For hot pressing large-diameter pieces, a high-frequency induction heating press is used. Successful applications include sections as large as 24 in. in diameter.

Applications: The hot-press method for the manufacture of large tungsten-carbide sections has been in use for some time. Improvements in technique have now made it practical to apply this method to a number of different sizes and shapes.

In this method graphite molds are machined from bar stock. The molds are usually the so-called "closing type;" that is, they are so dimensioned that the proper size piece has been produced when the plungers have compressed the carbide mixture so that the outer ends of the plungers are flush with the upper and lower surfaces of the die barrel.

From the known density of the end product, the weight of the desired carbide mixture is calculated and poured into the die cavity. The plungers are lightly pressed into the cavity, and the mold positioned in the hot press. The piece is hot pressed at relatively low pressure, on the order of 1000 psi. The final temperature is similar to the temperature used for sintering cold-pressed carbide, usually about 2450 to 2650 F depending upon the cobalt content. Care must be exercised not to underpress or overpress. If the carbide mixture is

not fully compressed, porosity will result in the finished carbide. On the other hand, overpressing will cause execessive squeezing out of cobalt binder, causing improper binder content, and local segregation of cobalt which again results in porosity or soft spots. For this reason, even with closing type molds, the compression of the carbide charge is carefully measured with dial indicators.

Mold Characteristics

Unfortunately, in most cases, it is impossible to salvage the graphite mold. However, there are many cases where this procedure is economically practical. On plain round cylinders and bushings, standard graphite round bar stock can be purchased in a number of sizes, so that often only simple cutting-off and drilling operations are required to produce the desired mold. On small pieces it is possible to use multicavity molds in which several pieces are made simultaneously. For shaped pieces, it is often easier to machine a graphite mold than it is to machine presintered cold-pressed carbide, which is very fragile and must be handled and machined with extreme care. One of the advantages of hot-pressed carbide is the absence of sinter shrinkage. It is well-known that cold-pressed carbides may shrink as much as 18 per cent lineally or 50 per cent by volume during the sintering operation. Thus it is a difficult problem to avoid distortion on odd-shaped pieces or long thin-walled section which may vary in density as cold pressed. On the other hand, the hot-pressed carbide completely fills the mold cavity as it compresses, and the dimensions are those of the mold cavity.

Thousands of drawing dies have been salvaged by hot repressing. In this method, the die is placed in a graphite mold with a core of the desired diameter, less several thousandths of an inch grinding allowance. The die is hot pressed in the mold and, as the carbide becomes plastic, it is compressed slightly, filling in the bore and los-

ing a little in height. As a result of this technique, carbide is taken out of the height, which is not critical, and transferred to fill in the bore, which is the critical dimension. The graphite molds can be used several times, and the surface of the carbide die retains a fairly good finish. As a result only a few thousandths need be ground off the bore surface to produce a high degree of accuracy and finish.

Dies for drawing tubing have been repressed as many as three times, and are still in use. The interesting part is that the die life after repressing has increased appreciably over that obtained on the original cold-pressed and sintered die. It is believed that this is due to the elimination of microporosity by the hot-repressing operation.

Wear-Resistant Matrices

Some interesting applications of the hot press technique are being made in the diamond dressing, grinding, and drilling fields. A good example is diamond mining bits, for exploratory, blast hole, and oil well drilling. Many such bits were formerly—and in some cases still are—produced by casting alloys such as beryllium-copper around diamonds. For most applications, however, manufacturers have been going to the hot-press method. Metal matrices are produced which resist wear and erosion, and retain the diamonds sufficiently well, so that they are salvaged and reused in new bits after their exposed points have been worn smooth. The diamond bits are being produced with a variety of metal matrices, ranging in hardness from about 20 Rockwell C to the full hardness of tungsten carbide. The particular matrix used depends upon the service to which the bit will be subjected and the type of formation in which the bit will be used. In similar fashion, hot-pressed metal-powder matrices are employed in the production of diamond dressing tools, metal-bonded diamond wheels, and other products.

A recent development is the use of the hot-press process for the

production of heavy metals. These heavy alloys have been produced for more than a decade by the cold-press and sinter process. Alloy composition is about 90 per cent tungsten with 10 per cent nickel-copper binder. Theoretical density ranges from 16.5 to 17.2 for the usual percentages of binder employed.

There are difficulties involved in the production of cold-pressed and sintered heavy alloys which are similar to those encountered in sintered tungsten carbide. If one desires to obtain close to theoretical density, very great care must be exercised in controlling particle size and purity of powders, sintering temperature, etc. During the sintering of cold-pressed pieces, shrinkage as high as 20 per cent lineally may take place. This makes it difficult, particularly on large sections, to control dimensions and to prevent distortion. For objects such as tall bushings, it is impossible to eliminate the "hour-glass" shape due to differences in density prior to sintering.

Coating Protects Molds

In developing the hot-press method for producing high-density materials, the usual graphite molds were used. However, one of the first problems ecountered was carburizing of the tungsten by the mold. A coating was developed, which, when painted or sprayed on the mold, not only prevents this carburizing tendency, but also protects the mold itself. Thus, a number of pressings can be made from one mold.

By the use of the hot-press method, it is relatively easy to achieve full density, and parts have been produced to densities ranging from 16.5 to 17.1, depending upon the particular composition. There is no problem of distortion, as the metal under heat and pressure completely fills the mold. The structure of the hot-pressed heavy alloy is much finer in crystal size than that of its cold-pressed sintered counterpart. This is because the hot-press cycle is relatively fast, whereas sintering to high density requires

sufficient time to cause considerable growth in the crystal size of the tungsten. Machinability of the hot-pressed alloy is considerably better than that of the sintered material, a result believed to be due to the difference in crystal structure.

Some sections produced by hot pressing formerly required machining away as much as 50 per cent of the tungsten alloy from the sintered piece to produce the required contour. It is anticipated that expansion of the hot-pressed heavy metal applications will result in considerable savings in tungsten scrap and machining time.

Variations of the hot-press technique are applied to other products. Beryllium has been hot pressed by using a high-frequency induction hot press, a graphite die with a chromium-plated steel or an Inconel liner, and low-carbon steel punches. A hot pressing of more than 30 lb in a 12-in. diameter die was produced. The properties compared favorably with vacuum-cast extruded material.

A form of hot-pressing, or sintering under pressure, is used in producing metallic friction materials. The friction surface or wafer is welded to a steel back and brought to final density by being sintered in a special bell type furnace, which applies pressure to a stack of the friction wafers and steel plates. Large silver base contact materials have for a number of years been produced by hot pressing.

New Developments: One of the virgin fields of research is that involving high-temperature alloy or metal-ceramic materials. Considerable time, money, and energy are going into powder metallurgy research involving high-temperature materials. In one program the objective was to see whether an alloy prepared by hot pressing powders would have properties similar to that of the same alloy prepared by forging. After considerable work, a method was developed whereby a hot-pressed alloy was prepared which showed the same high-temperature stress-rupture value as the forging. Subsequently, advantage of the powder metal process was taken, whereby certain materials and grain orientation were introduced which are impossible to obtain in material which begins as a cast ingot. In a surprisingly short time this technique resulted in a modified alloy which showed over five times the stress-rupture life at high temperatures of the forged material. Since then, a number of alloys and materials have been developed by the hot-press method. Many of these show unusual mechanical properties. Steels for more ordinary uses, both carbon and alloy, have been produced experimentally.

Of course, the problem of a suitable die material always rears its ugly head. Just about every type of ceramic material available, as well as high-temperature alloys, and cermets, have been tried. A permanent hot-press die material is yet to be found. However, there is another approach which shows considerable promise. In this method a cheap, discardable die is molded, the main concern being simply whether it will produce one good pressing. In mass production the cost of such a die should be measurable in cents rather than dollars.

From a paper entitled "Recent Practical Applications of Hot Pressings" presented at the Eighth Annual Meeting of the Metal Powder Association in Chicago, Ill., April 1952.

Educating Mechanical Engineers

By Neil P. Bailey

*Professor and Head of Mechanical Engineering
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THE present extent and rate of growth of technical knowledge in all the areas of mechanical engineering must be admitted to be beyond the mastery of any single individual and far beyond the capacity of students. A balanced mechanical engineering curriculum cannot, therefore, be one that covers all related areas of technology. Rather, it must be one which teaches students to gather facts, analyze situations and evaluate solutions in each general area of


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mechanical engineering.

Even after it is admitted that a mechanical engineering curriculum is to be constituted only of organized course sequences in each of the major areas, the situation is far from being resolved. At the moment and by virtue of its historical seniority, power development with its ramifications is usually over-specialized in most curricula. Not only does it receive more than its share of the available curriculum time but also far too much of a student's effort goes into a familiarization with the physical facts of the subject and not nearly enough time in gaining experience in solving engineering situations of a comprehensive nature.

Integration Needed

The second major area of mechanical engineering, the processes of conception, design, and manufacture of devices is usually so dispersed as to lose much of its possible effectiveness for students. They study at one time or another many details that are involved in evolving a product to fill a need but only in retrospect, if at all, are they able to put the pieces together and see the sequence of steps involved. This situation is rationalized by insisting that after all, designing is an art that can be mastered only by practice after graduation.

Between 1920 and 1940 the design aspects of such courses as power plant design, combustion engine design, and heating design disappeared. By 1950 only machine design remained and it had become restricted to the mechanics of the metal parts of machines. Simultaneously, separate and often completely independent courses in shop processes, manufacturing methods, and production procedures came into being. Now all products must be conceived for their function, be proportioned for their use and be manufactured for their market, and these three things are not independent considerations. Mechanical designers must project their minds into new

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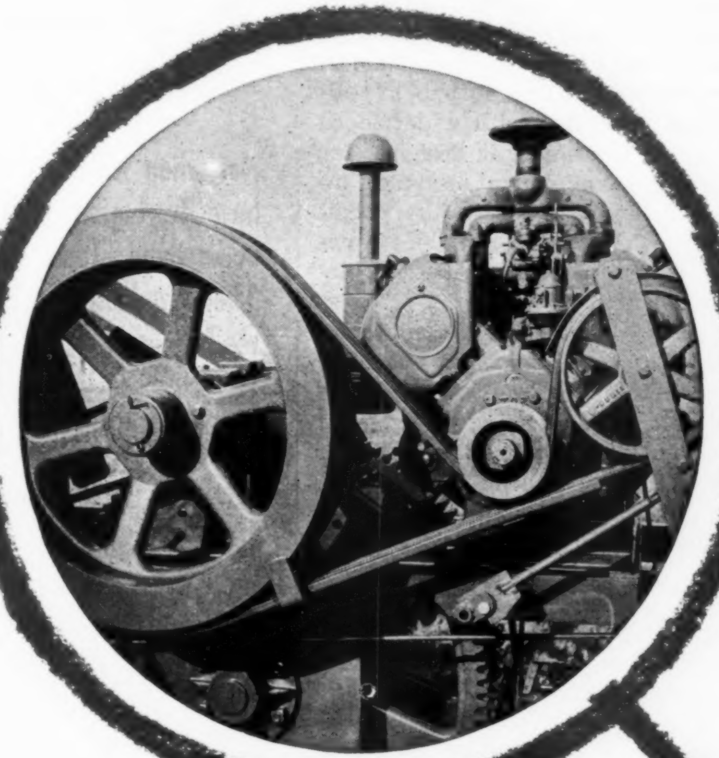
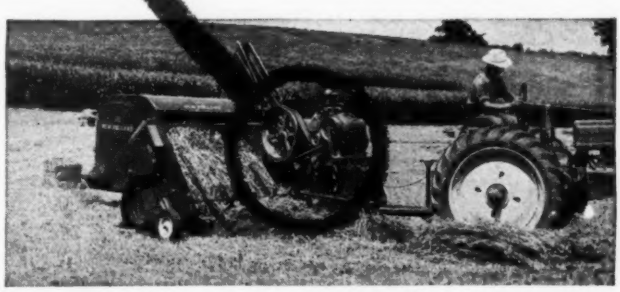




Baling is a one-man job with the New Holland Machine Company's Model 77 twine-tying automatic baler. Gilmer Multiple V-Belts contribute to its top performance.

Here's a noteworthy example of a machine designer's ingenuity ... and Gilmer V-Belts' versatility. The main drive on the New Holland Model 77 twine-tying hay baler is unusual in that it employs Gilmer V-Belts *running up-side-down!*

The multiple V-Belts drive off the small diameter sheave, while the *backs* of the belts operate over the large flywheel and the idler pulley. Ingenious designing!

Equally shrewd was the choice of GILMER V-Belts—engineered and built to take abnormal usage in their stride! It is unusual for V-Belts to be called upon to stretch and compress alternately on *both top and bottom.* But this two-way stretch puts little added strain on GILMER V-Belts. Their bias-woven jackets permit flexing in either direction. And their load-carrying, high-tensile-strength cords are centrally located—occupying the entire pitch-line area of the belt. All components are thoroughly bonded, so flexing in either direction creates a minimum of internal friction.

For greatest flexibility in designing, plan your drives around the broad range of GILMER V-Belt sizes. If special belts are required—or if you'd like assistance with your drive problems, NYB&P engineering, production and distribution facilities are at your service.

V-BELTS

NEW YORK BELTING & PACKING CO. Founded 1846

Our Second Century of Service to Industry

GILMER V-BELTS ARE NOW PRODUCTS OF

NEW YORK BELTING & PACKING CO.

America's Oldest Manufacturer of Industrial Rubber Products

1 MARKET STREET—PASSAIC, NEW JERSEY

situations and simultaneously weigh all of these factors. Regardless of the organizational changes that become involved, educators must find ways to develop design as a broad process of mechanical engineering. The output of talented graduates with a developed interest and ability in design is far below the minimum needs of industry and well below the potentialities of the American youth entering mechanical engineering departments.

As early as 1920 there was a growing conviction that engineering thinking did not include enough understanding of practical everyday relations with people in industry and business. Between then and 1950 the curriculum time aimed at this area of instruction was increased to as much as one-fifth of the total. Neither the mechanical engineering teaching departments nor engineers in industry are favorably impressed by the results realized to date because the original aim has not been achieved.

The undergraduate thesis has disappeared, and the number of project courses has decreased as the percentage of formal subject matter courses has become greater. Irrespective of the merits or faults of subject matter courses and old project courses this represents a trend away from the emphasis on the training of the individual student toward an emphasis on subject matter mastered. This, coupled with a steady decrease in laboratory and supervised design and problem periods should be reviewed thoughtfully. At a time when there is a recognized need for more originality, creativeness and individual ingenuity among mechanical engineers it would be sad indeed to be turning out a generation of students who could analyze and criticize ideas with great skill if only someone else would have the ideas.

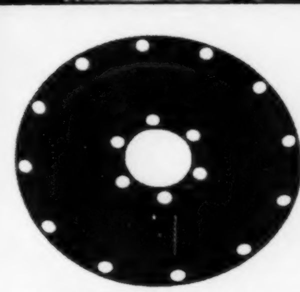
From a paper entitled "What are Contemporary Demands on the Engineering Curricula — Mechanical Engineering" presented at the joint ASEE-ECPD meeting in Chicago, Ill., September 1952.

How U. S. Rubber helped develop a stamping press

The new clutch assembly was ready to activate huge presses . . . except for one very important part, the diaphragm. U. S. Rubber engineers were called in and presented with the problem . . . a diaphragm that could stand up under all conditions and long continuous operation.

The answer was a diaphragm made of nylon cloth impregnated with Buna-N. When tested it exceeded the greatest requirements of the clutch assembly. One more example of how U. S. Rubber helps manufacturers everywhere develop new products to serve industry better.

Product of U.S. RUBBER — SERVING THROUGH SCIENCE



This diaphragm made over 6,000,000 cycles in the clutch assembly of a press such as those shown above. After this strenuous test, it shows no appreciable wear.

UNITED STATES RUBBER COMPANY

MECHANICAL GOODS DIVISION · ROCKEFELLER CENTER, NEW YORK 20, N. Y.

Review of Surface Finish Literature

By John W. Sawyer
Bureau of Ships
Department of the Navy

CONCLUDING the article "Review of Surface Finish Literature," which appeared in the September and October issues of MACHINE DESIGN, this subject index refers to the abstracts which comprised the first two parts of the article. Literature published during the period 1945 to 1951 on the subject of surface finish was abstracted.

—A—

—B—

—C—

check these advantages of *new*

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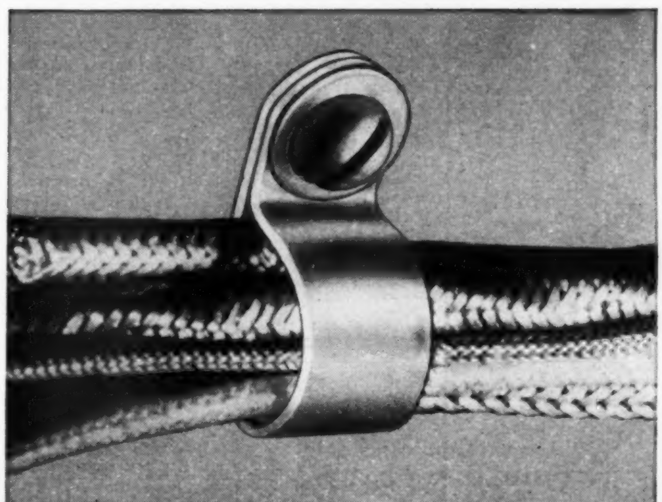
T.M.

nylon cable hangers

These new, tough, lightweight, pre-formed Nylon cable hangers, developed by Burndy research, give you the important features of metal *plus* these seven important *extras:*

- **Resistant to heat and cold** ... withstand sustained temperature from −60°F to 250°F.
- **Resistant to solvents**... undamaged by oils, gasoline, alcohol, hydraulic fluids.
- **High physical strength**... hold shape indefinitely — even under severe stress.
- **High dielectric strength**... can't short, nor ground. No hysterisis loss.
- **70% lighter weight**... than metal cable hangers of comparable size.
- **11 standard sizes**... to accommodate cables or bundles from 3/16" to 2".
- **Easier working**... clean, smooth, non-abrasive surfaces, rounded edges speed assembly, protect cable insulation.



Write for samples, prices, delivery information.

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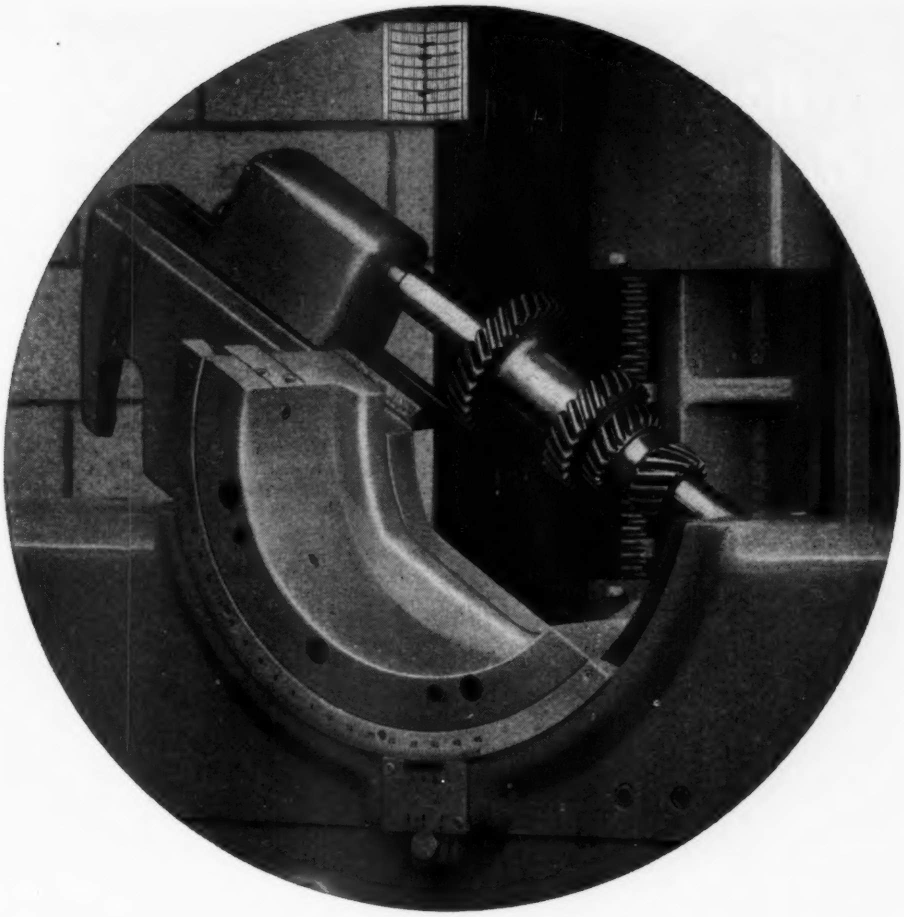
52-32

Surface Finish

here's a practical way to check spur and helical gears



The Kodak Conju-Gage Gear Checker automatically records the composite effects of runout, base pitch error, tooth thickness variation, profile error, lead error, and lateral runout. Illustrated is the Kodak Conju-Gage Gear Checker, Model 4U, for gears up to 4⅛″ pitch diameter. Larger and smaller models are also available.

In the final analysis, the accuracy of a gear is judged by whether or not it is good enough for the job.

Individual checks on errors such as profile error and pitch error can show tolerable variations which could combine to give a result intolerable in performance. Conversely, two intolerable errors might cancel out. But the composite check recommended in the American Standard (AGMA 236.02; ASA B6.11-1951) tests gears in action—tells at once whether or not the gear will do the job required.

With a Kodak Conju-Gage Gear Checker, you can check composite error to the closest tolerances. The reason lies in the Kodak Conju-Gage Worm Section. A single Worm Section, replacing a collection of master gears, can be used to check any spur or helical gear of any helix angle providing it has the same normal pitch and pressure angle. Its inherently simple and accurate form permits grinding to tolerances almost impossible to obtain in circular masters, especially in finer pitches.

Quick, conclusive, and accurate, the Kodak Conju-Gage Gear Checker makes it easy to meet specifications limiting tooth-to-tooth composite error to as little as .0002″. It automatically writes records to hold or ship with finished gears. And the gaging element can be checked in your own toolroom—economically reground to original specifications if necessary.

To find out how the Kodak Conju-Gage Gear Checker can solve your precision inspection problems, send for the booklet, "Kodak Conju-Gage Gear Testing Principle." It's yours for the asking by writing to Eastman Kodak Company, Industrial Optical Sales Division, Rochester 4, N. Y.

CONJU-GAGE INSTRUMENTATION

. . . a new way to check gear precision in action

To inspect all kinds of complex parts on a bright screen, Kodak also makes two highly versatile contour projectors.

Kodak

Surface Finish

Surface Finish

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—T—

—U, V, W, X, Y, Z—

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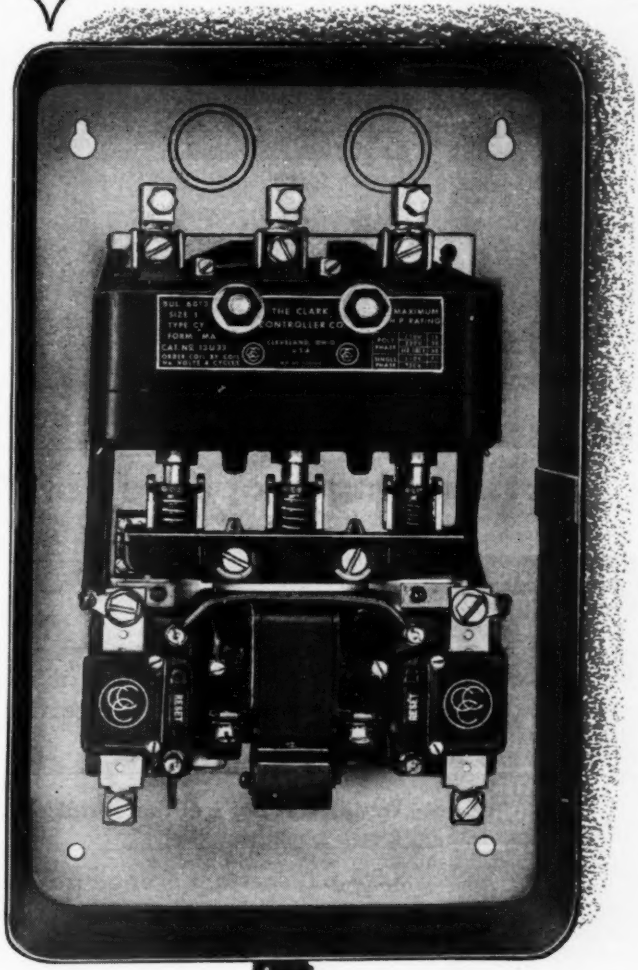
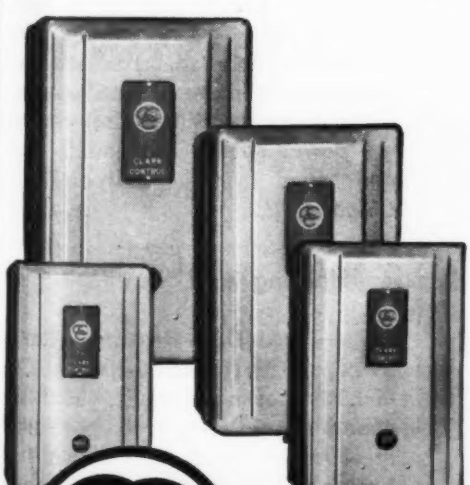
AC MAGNETIC MOTOR STARTERS

Now Available in SIZE 3

Now you can get this revolutionary starter in sizes 0, 1, 2, *and 3*. The size 3 uses the same new principle of arc interruption—with magnetic blow-outs and twin-break contacts—which has proven so successful on size 2 . . . Here is a rugged AC Magnetic Starter that provides the extra protection and dependable operation of Mill-type, heavy-duty construction in a compact unit. In addition to the many built-in features which mean better performance, this starter is designed to reduce maintenance costs to an absolute minimum. Contact surfaces can be quickly inspected without tools. Contacts can be quickly replaced with a screw-driver and a wrench without removing starter from cabinet. All parts are accessible and removable from the front. Captive screws are used wherever possible to simplify reassembly.



Clark Bulletin 6013 type "CY" AC Magnetic Starter, Size 3 with NEMA 1 cabinet cover removed. Note clean design, with plenty of wiring space.



CLARK Type "CY" AC Magnetic Starters are available in a wide variety of sizes, types of cabinets, and combinations of devices. Write for information.

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New Machines

Domestic

Food Freezers: Model HR-92 has 9.2 cu ft (322 lb) capacity; Model HR-132, 13.2 cu ft (462 lb). Counter height, flat top construction. Top is insulated with 5-in. layer of fibrous glass for greater effectiveness in retaining zero cold temperatures throughout interior. Freezing compartment separate from storage space is refrigerated on the sides and bottom so that foods can be frozen quickly. Equipped with sliding storage baskets and, on larger model, movable dividers to provide easier stacking of food. Interior light recessed in freezer top automatically lights storage compartment when top is opened. Cooling coils built into front, back and sides of the cabinet are spaced closer together at the top to equalize cabinet temperature. Size, 36 in. high, 31 in. deep, and 53½ and 70⅜ in. long respectively. *Frigidaire Div., General Motors Corp., Dayton, O.*

Automatic Dishwasher: Preplumbed MCP-14 "QuiKonect" model uses simplified drain unit that can be plumbed out either side of appliance base, depending on which side of existing plumbing dishwasher is installed. Has standard features including front opening, electric drying, single dial control, top power spray, and double wash and rinse cycles. *Hotpoint Co., Chicago, Ill.*

Heating and Ventilating

Gas-Fired Unit Heaters: Range in capacities from 50,000 to 230,000 Btu per hour input. Heat exchanger has staggered horizontal steel tubes which are rolled into heavy end sheets, boiler tube style, and fortified with additional outer and inner rolled flanges. Heat generator includes burner heads and mixing tubes of one-piece cast iron construction engineered for combustion of natural, manufactured, mixed or liquid petroleum gas.

FREE Cylinder Catalog and Templates offered by Ortman-Miller Co.

Book outlines several special features of O-M cylinders.

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A new, 28 page catalog has just been released by the Ortman-Miller Machine Co. which gives complete engineering specifications, data on O-M's special internal locking system and special full listing of O-M parts. Prepared for designers and users of cylinders for any application, it covers standard, oversize and 2-1 piston rods, giving full information on all sizes from 1½" to 8" bores.

SPECIAL FEATURES

Included are detailed explanations of the many features which have made ORTMAN-MILLER cylinders standard in thousands of plants throughout the country. Detailed drawings and copy explain the special shear bar assembly which completely eliminates bulky end caps and tie rods, thus saving up to ⅓ in space. In addition, it shows the vast number of interchangeable mountings and applications which almost always eliminate the need for special castings or patterns. This feature alone not only saves initial costs, but cuts down on inventory and greatly speeds up delivery on every order.

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For your FREE Catalog or templates, use the coupon in the ad at the right. Or write to ORTMAN-MILLER Machine Co., 1210 150th St., Hammond, Indiana.

New Machines

Burners, pilot and control valves are combined in removable drawer type assembly. Curved louvers direct heated air to worker zone. *The Trane Co., La Crosse, Wis.*

Room Air Conditioner: Small Model G3A for mounting on window sill. Main weight of unit balances on window sill; unit projects only 9 in. into room and does not extend outside normal building line. Cools rooms up to 350 sq ft at night and rooms up to 230 sq ft if used both night and day. Rated at 1/3-hp; operates on 115-v, ac line. *Quiet-Heet Mfg. Corp., Newark, N. J.*

Heater: Model VO-168 has adjustable vent for permanent installation or portable use. Operates on counterflow principle of spraying heated air out across floor areas to form a 6-ft high blanket of heat in work zone. Recommended for permanent use in confined areas such as small plants, garages, offices, warehouses, etc. Heat output vented is 130,000 Btu per hour. When used without a flue, free heat is directed back into blower and discharged "superheated," increasing output to 168,000 Btu per hour. Operates from detached fuel supply, burning No. 1 or No. 2 fuel oil, kerosene or diesel oil. Size, 21 by 33 by 69 in.; weight, 345 lb. *Fageol Heat Machine Co., Detroit, Mich.*

Materials Handling

Fork Lift Truck: Model 350, ½-ton capacity, for use in confined areas, gangways or upper floors. Fork height closed, 68 in.; open, 114 in. Overall length, without forks, 56 in.; width, 34 in. Turning radius, without forks, 55 in.; with forks, 80 in. *Hambro Machinery Div., The Powerad Co., New York, N. Y.*

Pneumatic Hoists: Redesigned line features safety load hook as standard equipment; safety, optional for suspension hook; improved lubrication system; extra bearing in motor drive shaft; heavier control level; hardened ring gears. Pendant controls available for handling unwieldy loads; ac-

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When requirements call for practically pure direct current or high cycle power for laboratory or production, inquiries invariably pin point in the Bogue direction. And, Bogue engineering-production ability has long been known for fine control equipment.

As we look forward to our 61st year of service to American Industry we pledge an ever increasing quality of power equipment to meet your varying needs for high precision products.

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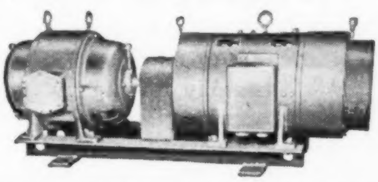

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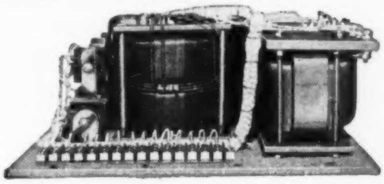
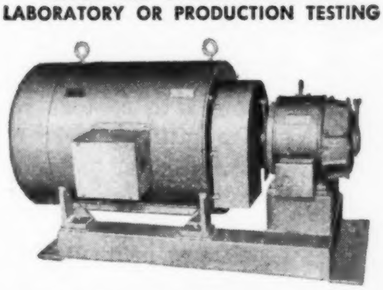
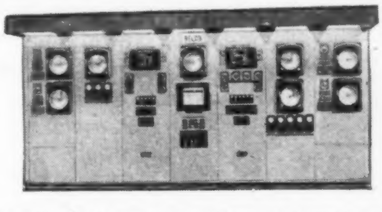


BOGUE 400 CYCLE POWER FOR LABORATORY OR PRODUCTION TESTING



PANELS FOR AUTOMATIC PROCESS CONTROL



AND . . . Precision Selenium Rectifiers, AC and DC Motors and Generators, Alternators, Power Supplies for Controlled Current and Voltage with output regulated to 1%—less than 1% ripple, Magnetic Controllers, Marine Equipment, Railway Equipment, Aircraft Equipment, Petroleum Equipment, Communication Equipment.

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BOGUE PRECISION POWER

cessories available include chain baskets, hose trolleys, I-beam hoist trolleys. *Keller Tool Co., Grand Haven, Mich.*

Fork Lift Trucks: Two gasoline-powered models with capacities of 8000 and 6000 lb. Features include long-life clutch that can be removed or installed in less than an hour and large heavy-duty disk type industrial brakes. *Hyster Co., Portland, Ore.*

Turn Tables: Composed of inverted swivel casters which permit moving and turning material for transfer between conveyors or machines, fabrication of sheets, assembly of heavy units, packing, marking, nailing and sealing of large containers, etc. Caster wheels are of neoprene or other synthetic material for protection of finished surfaces, or of semisteel for other applications. Caster swivels have double ball races. Wheels of light and medium duty tables have oilless bearings; wheels of heavy duty tables have hardened and ground steel bushings. Furnished in any required size, with or without floor supports. *Samuel Olson Mfg. Co. Inc., Chicago, Ill.*

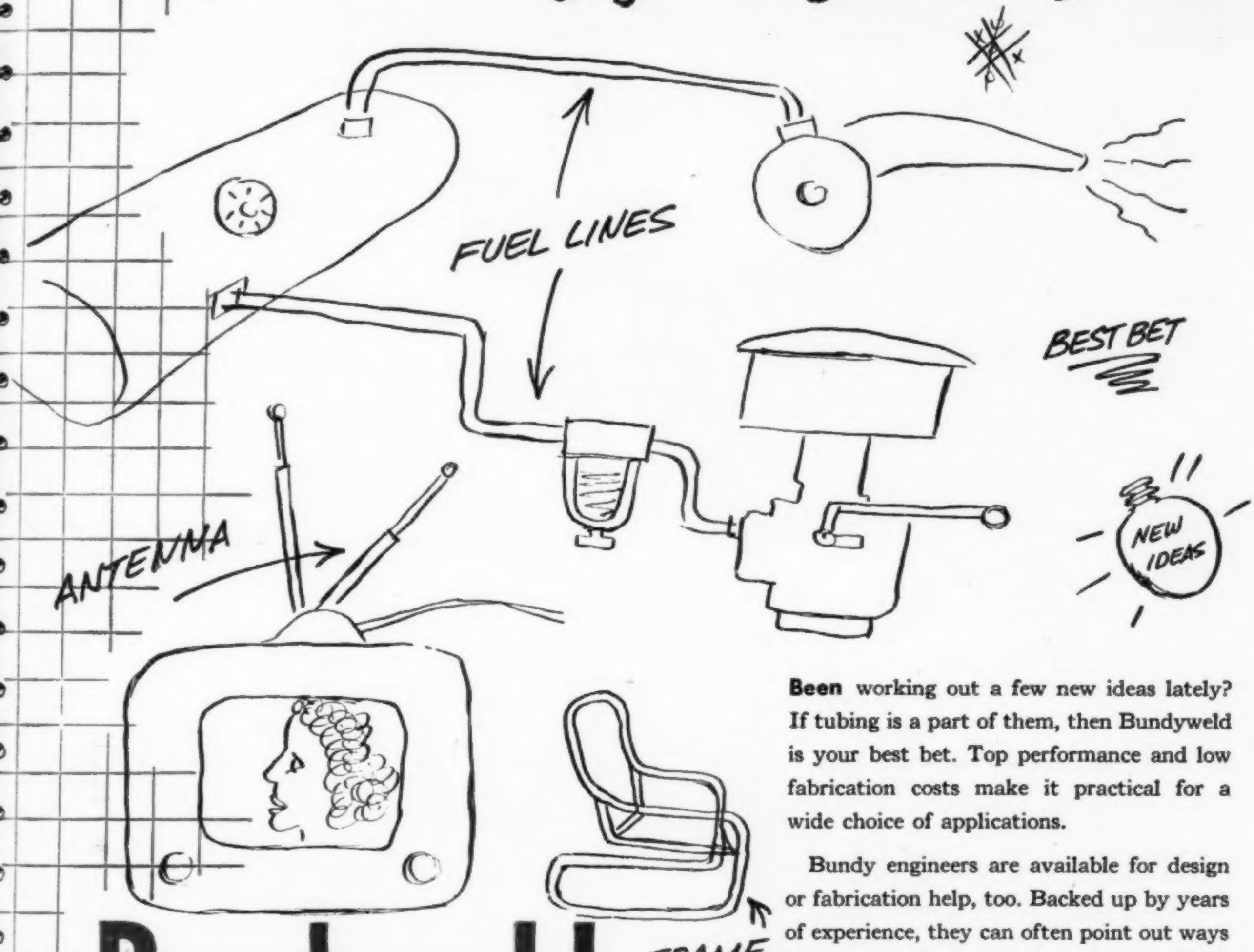
Overhead Electric Cranes: Range from 1 to 20 tons capacity. Feature antifriction bearings throughout, rotating axles on both bridge and trolley, variable-speed magnetic control operated by pushbuttons on floor controlled cranes and by master switches in a new pulpit type cage on cage controlled cranes. All gears operate in oil in sealed housings. Available with wide variety of speeds and heights of lift. *Shaw-Box Crane & Hoist Div., Manning, Maxwell & Moore Inc., Muskegon, Mich.*

Metalworking

Tapping, Drilling Machine: Electric, air controlled unit for use with either multiple or single spindle drilling and tapping heads, with provision for interchangeable work holders. Features electrically controlled four-way air valve with built-in spindle travel speed control. Can be run on continuous automatic cycle or single stroke operation controlled by foot or

"Bundyweld "Doodles"

to jog a designer's imagination



Been working out a few new ideas lately? If tubing is a part of them, then Bundyweld is your best bet. Top performance and low fabrication costs make it practical for a wide choice of applications.

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DOUBLE-WALLED FROM A SINGLE STRIP

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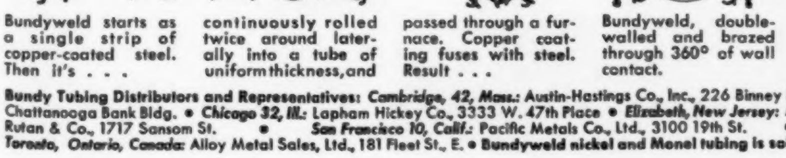
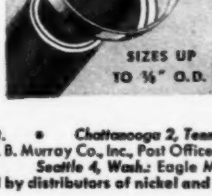
- **Leakproof**
- **High thermal conductivity**
- **High bursting strength**
- **High endurance limit**
- **Extra-strong**
- **Shock-resistant**
- **Ductile**
- **Lightweight**
- **Machines easily**
- **Takes plastic coating**
- **Scale-free**
- **Bright and clean**
- **No inside bead**
- **Uniform O.D., I.D.**

WHY BUNDYWELD IS BETTER TUBING



Bundyweld starts as a single strip of copper-coated steel. Then it's . . . continuously rolled twice around laterally into a tube of uniform thickness, and passed through a furnace. Copper coating fuses with steel. Result . . . Bundyweld, double-walled and brazed through 360° of wall contact.



← **NOTE** the exclusive patented Bundyweld beveled edges, which afford a smoother joint, absence of bead and less chance for any leakage.

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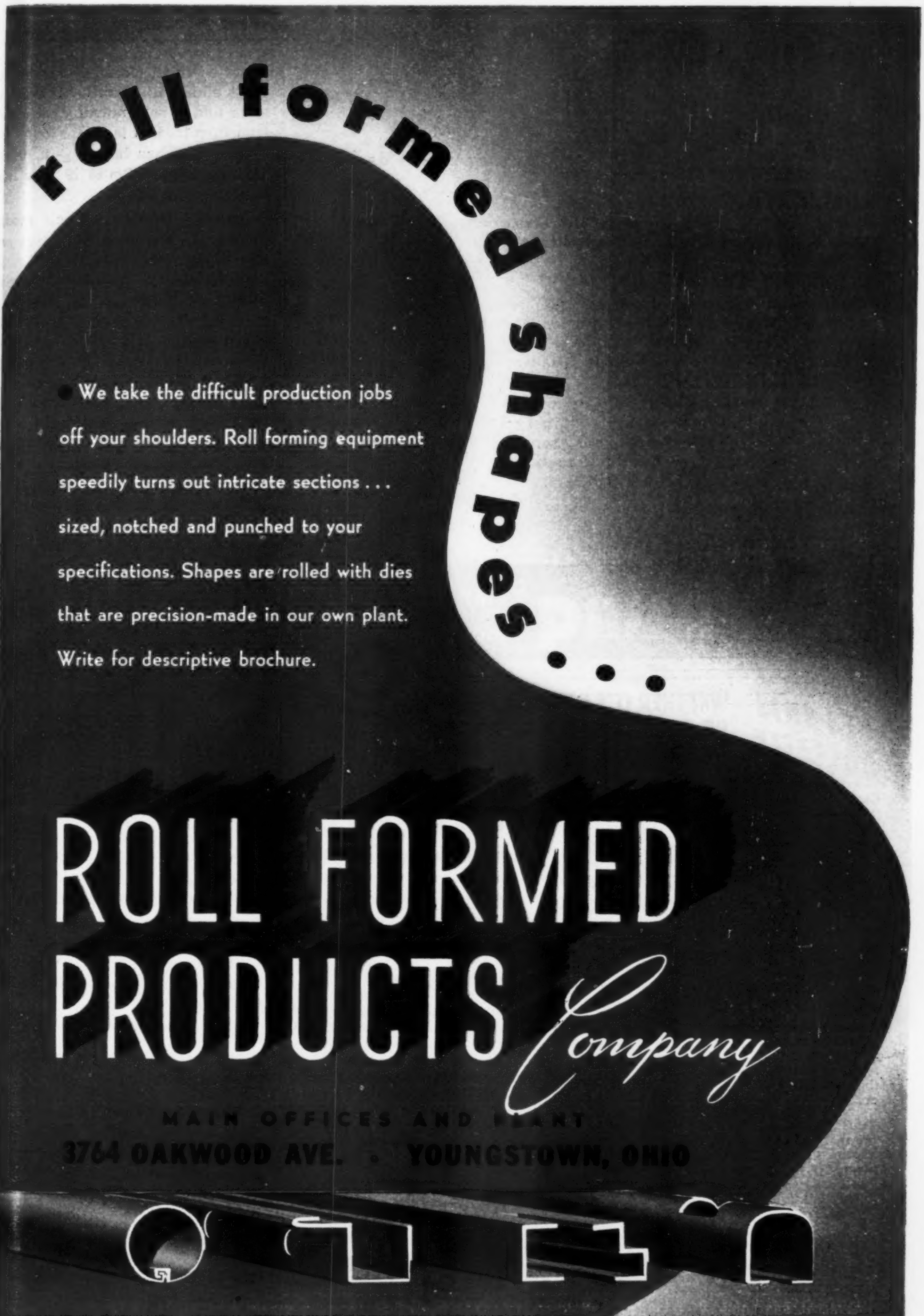
hand. Single spindle tapping attachments which take taps up to ¾-in. are available from stock. *Ettco Tool Company Inc., Brooklyn, N. Y.*

Rotary Table: Diameter, 12 in. Has four longitudinal T-slots. Top is graduated from 0 to 360 degrees; dial is graduated in minutes. Has an adjustable marker for fast setups and an eccentric cam which disengages the worm for production indexing; also has an adjustable stop for positive return and wear take-up. Accurate to within 0.0005-in. in concentricity and 0.001-in. in parallelism with base. Has 7¾ in. diameter hold-down ring and free access to pilot hole from the underside for center clamping or pulldown. *Kenco Mfg. Co., Los Angeles, Calif.*

Thread Roller: Model 300 can be used on either hollow or solid work in various metals ranging from nonferrous, through aluminum and magnesium, to hardened and stainless steels. Rated at 19,440 pieces per hour of external, class 3 fit threads rolled in ½-in.—13 hollow set screws of SAE 1035 steel. "Planetary die" principle distributes thread rolling pressure evenly over 30 in. of die. Pieces are hopper fed and roll at close intervals around circumference of die. Large number of pieces simultaneously in work makes possible slower die speeds. *D. H. Prutton Machinery Co., Cleveland, O.*

Radial Arm Machine: Single-horsepower, for use in small workshops or in the fabrication of light metals. Makes straight cuts, miters, bevels and compound-miter cuts. Can be used with metal cutting saw blades on aluminum and other nonferrous stock or with abrasive wheels on small-size steel and other ferrous metals. By replacing the metal-cutting blade with a wood-cutting blade, the machine can be converted for normal radial arm woodworking operations. *DeWalt Inc., Lancaster, Pa.*

Instrument Turret Lathe: Small, sensitive bench model designed to handle very small parts. Made with collet capacities of 3/16 or 5/16-in. Equipped with a lever-



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New Machines

operated collet closer, a double tool cross slide with a swivel compound slide and a self-indexing, six-position turret. Bed is 18 in. long, and turret head has holes ½-in. in diameter. *Louis Levin & Son Inc., Los Angeles, Calif.*

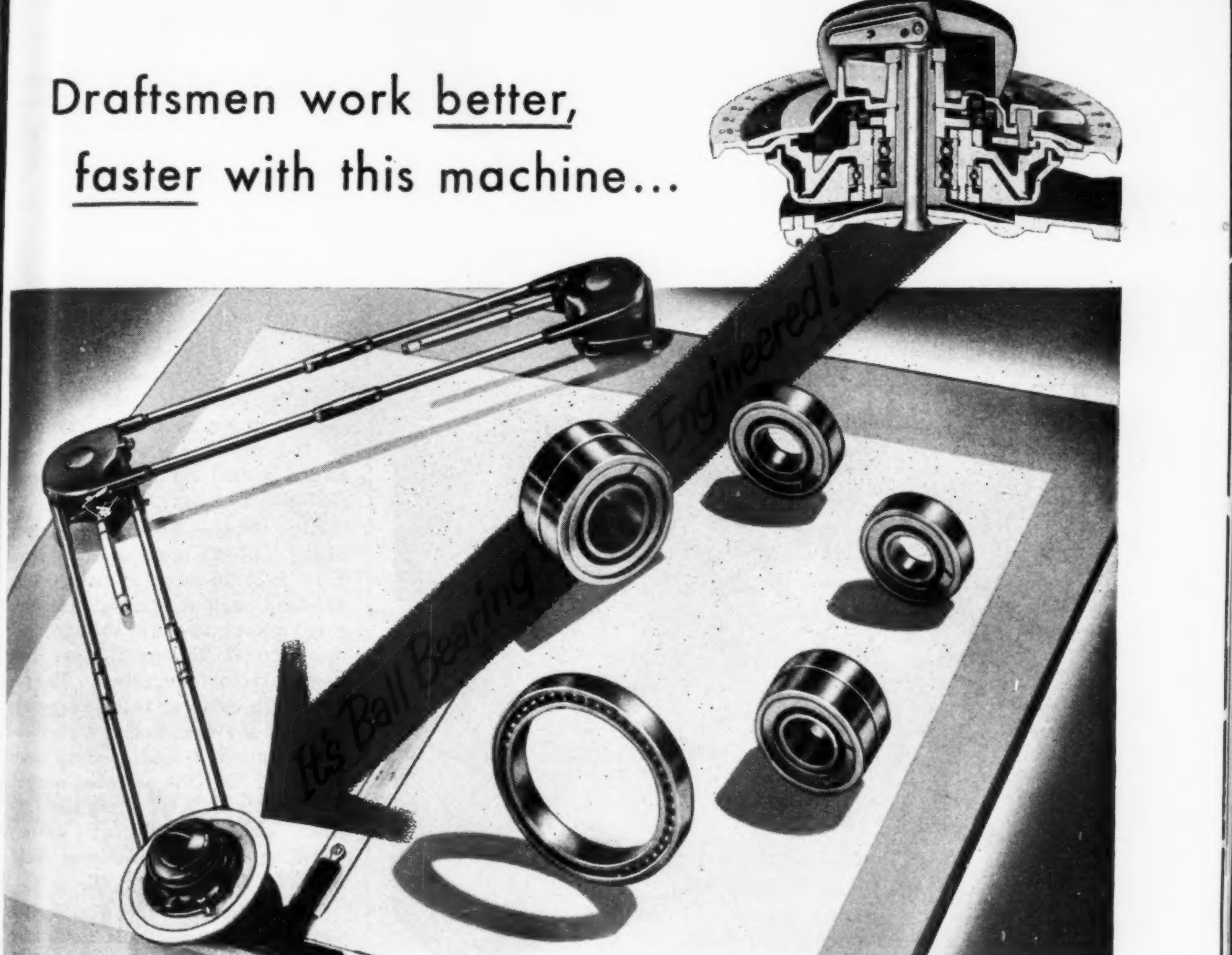
Bending Brake: Bench model, hand-operated, universal, box and pan brake for bending sheet metal and steel plate. Bends sheet metal up to 18 gage and 24 in. long. Bending edge is sectional in graduated widths fitted to a bar. Sections can be adjusted or removed as work requires and can be used in any combination anywhere along the bending edge for folding, box and pan work, and a variety of straight bending operations. Positive, cam-action clamp adjusts for different thicknesses of material. Minimum size of reverse bends, ¼-in.; maximum angle of bends, 135 degrees; maximum depth of box or pan, 3 in. *Dries & Krump Mfg. Co., Chicago, Ill.*

Portable Pipe Threader: Has power-operated four-jawed chuck which eliminates use of wrenches or other manual tools in handling pipe. Can be used for making up, as well as threading, cutting and reaming. Has pipe or conduit range of ⅜ to 2 in. and bolt range of ⅜ to 1¼-in. Motor is 115-v ac or dc on any cycle. Other features include quick opening die heads; cutter, which is removable for short nipples, mounted on front jaw housing; and flat blade reamer which automatically reams while threading progresses. For bolt threading, reamer can be removed. *Quijada Tool Div., Gaines-Collins, Los Angeles, Calif.*

Plant Equipment

Monotube Dryer: Functions include drying, cooling and recovering by-products and solvents. Operates on conduction principle of heat transfer. Available in two trough diameters—24 in. in lengths from 5 to 10 ft, and 30 in. in lengths from 5 to 20 ft. For treating chemicals, pharmaceuticals, food and vegetable oil mill products, etc. Operates at high or low

Draftsmen work better, faster with this machine...



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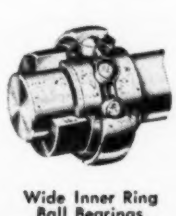
By holding spindles absolutely rigid and keeping motion friction-free, Fafnir Ball Bearings contribute much to the vernier accuracy and effortless action of the Universal "Boardmaster". The ultimate results are more accurate sketches, layouts or detailing and "on the nose" template work in faster time . . . so important when good engineers and draftsmen are scarce.

At three pivot points, pairs of extra-light, compact single row, Fafnir Ball Bearings give finger touch ease of movement without a trace of play. At the "centralized control unit", a large diameter Fafnir Bearing is used together with paired, preloaded and pre-lubricated Fafnirs. Six of the seven Fafnir Bearings used are Plya-Seal Type, which prevent lubricant leakage, keep out contaminants, and are lubricated for life.

The faculty for providing custom-made types of bearings out of stock . . . from the most complete line in America . . . makes Fafnir the first source consulted by leading manufacturers. Maybe you can solve bearing problems this easy way. The Fafnir Bearing Company, New Britain, Conn.

Standard Radial Bearings



Wide Inner Ring Ball Bearings with Mechani-Seals

Plya-Seal Type Bearings

FAFNIR
BALL BEARINGS
THE MOST COMPLETE LINE IN AMERICA

New Machines

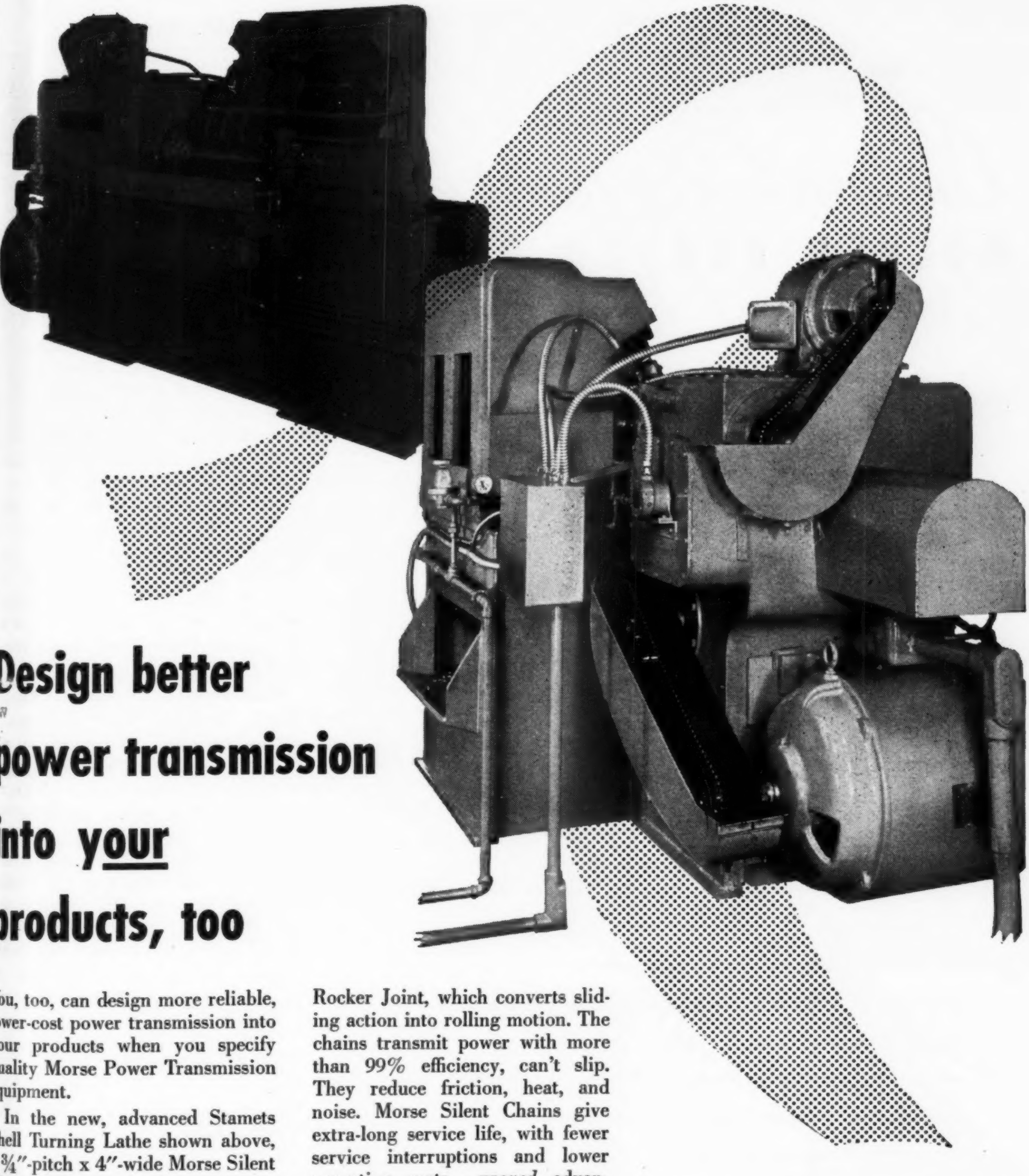
temperatures. Light, fluffy materials such as cake flour and cottonseed meal can be dried. Can be used for products which must be dried at temperatures above or below that of steam, by using another medium such as hot water for lower temperatures and hot oil or other liquid for higher temperatures. Can also serve as a cooler by using water, refrigerants or other cooling media, or, by the process of condensation, can extract by-products and solvents. Solvent in wet solids is carried out of the dryer by warm air and cooled beyond the dew point. *Link-Belt Co., Chicago, Ill.*

Arc Welder-Battery Charger: Model 175-BC handles rods from 1/16 to 3/16-in. Features 15 to 175-amp output range, 140-amp rated output, 25-v arc voltage, 65-v open circuit voltage, 75 per cent power factor correction, 50 per cent duty cycle at rated amperage load, 20 per cent duty cycle above rated amperage load, 36-amp draw at 220-v, 7.9 KVA at rated output, 6 dc amp battery charging rate. Also available without battery charging circuit. ***Mid-States Welding Mfg. Co., Chicago, Ill.***

Engine-Generator: Model 4500 operates many portable tools such as cut-off saws, electric chain saws, masonry saws, concrete vibrators or lights. Designed for full output at 115-v, 60-cycle ac power. Has motor starting capacity of 4500 w, which is sufficient to start and run motors up to 1½-hp. Intermittent rating is 3000 w; continuous rating is 2500 w. Available with either standard tubular cradle base or portable base attachment consisting of a single handle and semi-pneumatic rubber tired wheels. Designed for maximum portability with either a Briggs & Stratton No. 23 or Wisconsin AEN four-cycle engine. *Wincharger Corp., Sioux City, Ia.*

Processing

Roller Coater: Equipped with three sets of rolls; has automatic infeed and off-bearing conveyors. Rubber nip rolls convey



Design better power transmission into your products, too

You, too, can design more reliable, lower-cost power transmission into your products when you specify quality Morse Power Transmission equipment.

In the new, advanced Stamets Shell Turning Lathe shown above, a ¾″-pitch x 4″-wide Morse Silent Chain Drive functions as primary drive from 75-HP motor. The small carriage drive is a ½″-pitch x 1″-wide Morse Silent Chain Drive. Both drives are the result of Morse engineers working closely with Stamets engineers during lathe's design phase.

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New Machines

workpieces from infeed conveyor through a set of brush rolls for cleaning and then through a set of hardened steel coating rolls. Doctor rolls, controlled by a calibrated handwheel with micrometer adjustment, gage the thickness of the compound coating. Pressure on the coating rolls is applied by two 6-in. diameter air cylinders. Pressures from 0 to 160 lb can be achieved. Pressure behind the hardened steel coating rolls provides a deburring action on workpieces and at the same time presses the compound into the pores of the metal. Variable-speed coating rolls adjust for stock thickness from 30-gage to ½-in. and will take widths from 1 to 112 in. A curculating pump provides a continuous supply of compound to coating rolls. Machine can apply compound to one or both sides of the metal. *Union Tool Corp., Warsaw, Ind.*

Heat-Treating Furnace: Model FG-76 designed for continuous duty. Performs efficiently at all heat levels up to 2500 F, with higher temperatures available for short or intermittent runs. Constructed of heavy, gas-tight, electrically welded steel. Element parts are gasketed and tightened by means of thumb screws. Gas valve is attached to door pedal and actuated by its movement so that the flame curtain ignites as the door is opened and turns off when the door is closed. Can be adapted for use in reducing as well as oxidizing atmospheres. Chamber size, 8 in. wide by 16 in. deep by 6 in. high; overall size, 33 in. wide by 40 in. deep by 70 in. high. *Pereny Equipment Co., Columbus, O.*

Testing and Inspection

Electrical Comparators: Each one of new line provides four gaging ranges. Utilize electromagnetic principle of magnifying the movement of the spindle tip. Spindle pressure is adjustable from 4 to 40 oz. Spindle lifter is supplied to facilitate checking of very small or delicate parts. Gaging head swivels 360 degrees in both horizontal and vertical planes and can

New Machines

be removed for use in special gaging setups or in remote indication setups. Standard models have gaging capacity of 6½-in. maximum height and 4-in. depth. Special long column available which provides effective checking height of 20 in. Large three and four-point, and multiple V-anvils, back stops and special flat and long spindle tips are available for production inspection. *The DoAll Co., Des Plaines, Ill.*

Ultrasonic Micrometer: Measurements within 0.5 per cent of actual thickness can be obtained from one side of materials with smooth surfaces. Instrument can be used on homogeneous materials such as steel, brass, aluminum, copper, glass, lucite and other dense substances capable of transmitting the ultrasonic waves generated by a vibrating crystal. Measures thickness by determining frequency at which resonance of high-frequency sound waves occurs in the material under test. Resonance is indicated by maximum deflection of a meter and maximum strength of a tone audible in the headphones. Designed to operate with high precision over infinite series of relatively narrow thickness ranges between 0.010 and 12 in. Plug-in frequency coils and matched dials for desired frequency permit selection of specific frequency best suited to required thickness range. *Branson Instruments Inc., Stamford, Conn.*

Meter Tester: For testing and calibrating dc instruments. Measures internal resistance of sensitive instruments without exceeding full-scale rating of the instrument under test. Has regulated power supply, stepless vacuum tube voltage control, illuminated 8½ in. mirror-scale, hand-calibrated standard instrument, decade of 0.1 per cent accurate Manganin wire wound resistors. Range, 25 microamps full scale to 10 milliamps full scale, and 0-100-v. Overall accuracy is better than 0.25 per cent. Resistance range, 0-5000 ohms. Operates on power source of 115 v, 60 cycles, ac. Size, 15⅛ by 10⅛ by 5⅜ in.; weight, 15 lb. *Marion Electrical Instrument Co., Manchester, N. H.*